THE DESIGN OF

VALVE GEARS

FOR

STEAM ENGINES

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PREFACE

Of the many different "diagrams" which are used for investigating valve motions, nearly all are useful for the purpose of analyzing the action of existing gears, but most of them are unsuitable for determining the proportions of new ones. For the latter purpose the Bilgram Diagram undoubtedly possesses marked advantages over the others. Unfortunately, however, it is a little difficult for the beginner to understand and does not show how the valve openings vary throughout the cycle as clearly as do most of the other diagrams. Usually it should be employed for designing only, and then it is often advisable, at least for the beginner, to analyze the results obtained by it, by constructing one of the other simpler diagrams. For the purpose of analysis the Zeuner and the Elliptical Diagrams are generally considered to be the best. If one expects to have much to do with valve gears, he should have a thorough understanding of these three diagrams at least.

Needing a text-book for his own students, and finding among the many existing works on valve gears, none which employs a combination of the Bilgram, Zeuner and Elliptical Diagrams, has led the author to attempt to supply the want.

In preparing this work most of the standard books on the subject have been consulted, and much of the text is necessarily merely a new treatment of old matter. For that part relating to the Bilgram Diagram acknowledgement is due to Mr. F. A. Halsey, whose excellent treatise, "Slide Valve Gears," is the exponent of this diagram; and to Mr. E. T. Adams, who published some blue print notes on the subject some years ago.

There is some original matter, and some that has been collected from scattered scources and which has not been embodied in text-books before. For a great deal of this new matter the author is indebted to Professor John H. Barr, formerly Professor of Machine Design in Cornell University. Credit for matter obtained from other sources is given in the text.

The treatment is from the graphical standpoint, instead of from the mathematical, and might properly be termed the "Kinematics of Valve Gears." It is assumed that the student is already familiar with the action of the simple valve gear with the D-valve, and it is intended that the study of the text shall be accompanied by supplementary talks by the instructor and by a drawing board course, without which it is practically useless to attempt to study the subject. Throughout the text will be found questions for mental solution, and problems which require a drawing board treatment. The book is so arranged that there is space for the student to add notes of his own.

This opportunity is taken to thank Messrs. R. B. Renner and L. Illmer, Jr., for reading parts of the proof and for making valuable suggestions.

Ithaca, N. Y., Aug. 21, 1905.

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Part I

CHAPTER I.

THE PLAIN SLIDE VALVE—DEFINITIONS AND ACTION.

1. **INTRODUCTION.** This subject is of such a nature that it requires something more than the mere reading and studying of text to give one a working knowledge of it. The author believes that the only way to master the principles is for each individual himself **to construct the various diagrams and figures** as they are explained in the text, (often free hand pencil sketches are sufficient for this), and to **solve the various problems and to answer the questions** as he proceeds.

An extremely valuable asset is the **ability to picture in one's mind** the relative positions of the various parts of the mechanism when the position of one part is known. If the student will make a special effort at the outset to develop this ability, and will afterwards make constant use of it, he will find that he has a great advantage over those who have not taken the trouble to acquire it.

It is assumed that the student is already familiar with the arrangement and operation of the simple steam engine having the plain slide valve. The purpose of this first chapter is mainly to review certain definitions, to bring out certain conceptions, and to give the symbols, abbreviations and letters of reference which will be used through the text, so as to ensure a common basis of understanding before proceeding with the development of the subject.

Unless it is stated to the contrary it will be always understood that the engine is horizontal with the cylinder to the left, that an external D-valve is used and that the crank rotates clockwise.

2. **THE ENGINE.** A horizontal engine is said to be **running over** if the crank pin moves away from the cylinder when the crank is above the horizontal center line. It **runs under** when the crank is rotating in the reverse direction. In the first case the thrust of the connecting rod will cause the cross head to always press downward against the lower guiding surface; while in the latter case the pressure of the cross head will be upward. Engines are usually run over.

The **crank end** (C. E.) of the cylinder or valve is the one nearest the crank or next to the engine frame. This end is also called the front end. The opposite end is the **head end** (H. E.), sometimes called the back end.

The **forward** (Fd.) **stroke** of the piston or valve is that towards the crank. The return stroke is the **back** (Bk.) **stroke.**

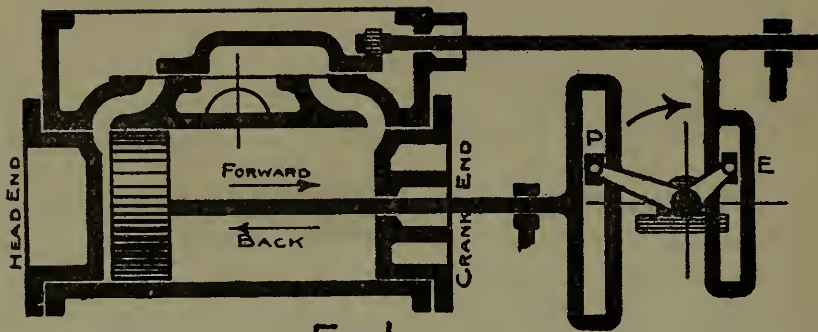


Fig. 1

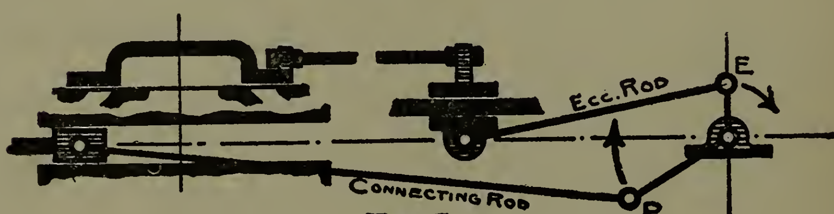


Fig. 2

Fig. 1 shows a skeleton outline of a simple valve gear having a **Scotch yoke** or **slotted crosshead.** As the engines are usually arranged, the back of the valve, instead of the section or edge, would be seen in this view, but for the purposes of analysis it is permissible to consider the valve as turned on its stem to the position shown here.

Fig. 2 shows the elements of the more usual form of gear, that having an eccentric rod. If the eccentric rod is of infinite length the motion of the valve will be the same as that which would be derived for a slotted crosshead. Such a rod will be spoken of as an "**infinite rod,**" to distinguish it from a "**finite rod,**" which is of finite length.

The crank is on **dead center** when the piston is at the end of the stroke, and is therefore horizontal on horizontal engines. When the piston is at the head end of the cylinder the crank is on the **head end dead center**; when at the other end, the crank is on the **crank end dead center.**

The **eccentric** (ecc. or E.) is really a crank pin of such large diameter as to surround the shaft. In the discussion of the action of the valve gear we are only interested in the motion of the center of the eccentric, to which the term will be applied hereafter. Being a crank, the eccentric has dead center positions.

The **throw** of the eccentric is the "eccentricity" or length of the crank. (There is a lack of agreement in the use of this term, some using it in the sense given and others as meaning the total movement of the valve or "travel.")

3. THE VALVE.

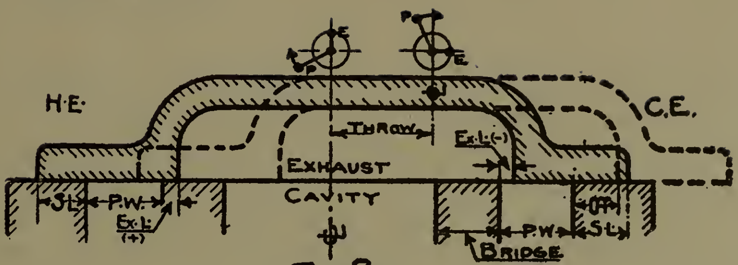


FIG. 3

In Fig. 3 is shown a longitudinal section of a valve seat and "D-valve," the latter being arranged to admit the steam at the ends and to exhaust at its middle.

The **valve seat** is the face of the cylinder on which the valve slides.

The openings in the valve seat for admitting steam to, or for exhausting it from, the cylinder are called respectively **steam** and **exhaust ports.**

The **passages** conduct the steam from the ports to the cylinder, or the reverse, and are termed **steam** or **exhaust** passages according to their function.

In engines having the simple valve the same passages and ports are used for both operations.

The **widths** of port, of passage and of valve opening are measured longitudinally, and the **lengths,** transversely, with respect to the cylinder. (In Fig. 3 the port width is p. w.)

When the simple D-valve is used, the widths, lengths and areas of port and passage are usually the same, so the terms "port" and "passage" are often considered as being synonymous. However, when other types of valves are used the widths (and consequently the areas) are often not the same, and it is then necessary to make a distinction in the use of these terms.

The **exhaust cavity,** or chamber, or chest, is that space into which the exhaust steam flows after leaving the port.

The **bridge** is the wall which separates the port from the exhaust cavity.

The **valve is central** when it is in the middle of its travel, or stroke, the eccentric being vertical either up or down.

The **valve** face is that surface of the valve which rests on the valve seat.

The **steam and exhaust edges** of the valve or of the port are respectively those which open to admit steam to, and to exhaust it from the cylinder. The **outer** and **inner edges** of the valve are respectively those at the ends and those toward the middle of the valve. Similarly, the outer and inner edges of the ports are

respectively those edges farthest away from and those nearest to the center of the valve seat.

The **lap** is the distance the edge of the valve is from that edge of the port with which it operates and is measured **when the valve is central.** The **outside lap** (or outer lap) is that of the outer edge and the **inside lap** is the lap of the inner edge of the valve. The **steam lap** (S. L.) and the **exhaust lap** (Ex. L.) are respectively those of the steam and exhaust edges of the valve. The lap is **positive** if the port is closed and **negative** if it is open. (Negative lap is sometimes called "clearance.") In Fig. 3 the exhaust lap at the crank end is negative. Similar laps at both ends of the valve need not necessarily be the same in amount.

The **valve opening** is variable but the term is usually employed as referring to the **maximum** opening.

When the movement of the valve is more than enough to open the port the excess movement is termed the **over travel.** (O. T. in Fig. 3.)

The **width of valve face** is the distance between the inner and outer edges of the same end of the valve and is therefore equal to the sum of the steam and exhaust laps and the port width.

The valve is said to **"take steam from the outside or ends,"** and is sometimes termed an **"external valve,"** when the steam enters the cylinder past the ends and exhausts at the middle, as in Fig. 3. Then the outside lap is the steam lap and the inside lap is the exhaust lap.

When a valve **"takes steam from the inside or middle,"** it is an **"internal valve."** Then, the inside lap is the steam lap, and the outside is the exhaust lap and the steam and exhaust chambers change places.

The **travel** or **stroke** of the valve is the amplitude of its motion.

When there is no rocker arm between the eccentric and the valve, arranged to multiply or reduce the motion, the travel of the valve is equal to the diameter of the eccentric circle.

The term **displacement,** when applied to the valve or piston, will be understood to mean the distance the center of the part has been moved from its **central** position; and in the case of the eccentric and crank it will be the **horizontal** distance from the center of the pin to the **vertical center line** of the shaft.

If the valve gear having the Scotch yoke (Fig. 1) is used, it is evident that the displacements of the valve and of the eccentric are equal, and that the valve has simple harmonic motion. If, however, the mechanism having an eccentric rod of finite length is employed, these displacements will not be equal (except when the eccentric is on dead center) and the motion of the valve will not

be simple harmonic, because of the "angularity" of the eccentric rod. The differences between the displacements are usually small, however, and when the analysis of the action of the valve is to be only approximate, it is often convenient to ignore the inequalities altogether and use the displacements given by the Scotch yoke. In this case the displacements and positions of the valve will be termed **nominal.**

What is true of the valve and eccentric displacements is, of course, also true of the displacements of the piston and crank.

For the present, only the case of the Scotch yoke will be considered. The effect of the angularity of the rods will be taken up in a later chapter.

> **X1.** (a) Lay out a symmetrical valve seat, with widths of exhaust cavity 5 inches, of bridges 1½ inches, and of ports 1¼ inches.
>
> (b) Draw in its central position an external valve having steam and exhaust laps respectively 1¼ inches and negative ⅛ inch for the head end, and 1⅛ inches and (positive) ⅛ inch for the crank end; thickness of metal 1 inch. Dimension and label completely.
>
> (c) If the throw of the eccentric is 2½ inches, what are the overtravels of the steam and exhaust edges, and what are the maximum steam and exhaust openings, of both ends of the valve.
>
> **X2.** Same as X1, but for an internal valve.

4. THE ACTION OF THE D-VALVE. Looking at Fig. 3, it will be seen that, for the steam edge of the valve to be just opening, the valve, and consequently the eccentric, must have displacements equal to the steam lap, in which case the eccentric will have rotated past its vertical position, through the angle λ, called the **lap angle,** as shown in Fig. 4 for the head end of the valve.

When the crank is on dead center, (the piston being at the end of the stroke) the steam edge of the valve is usually arranged to be open a small amount, which is called the **lead.** Then the valve and eccentric will be displaced a distance equal to the lap plus the lead (Fig. 4), and the angle through which the eccentric will have rotated past its vertical position is called the **angle of advance,** a. The angle through which the eccentric rotates while the valve is opening to lead is the

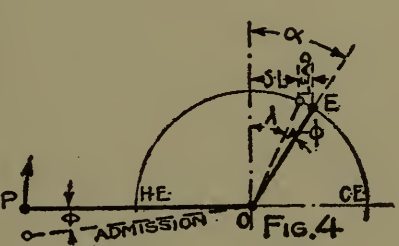


FIG. 4

lead angle, ϕ which is equal to $(a—\lambda)$. The exhaust edge also has angles of lead and lap, but these terms are always understood to apply to the steam edge, unless otherwise stated.

It is evident that when the throw, lap and lead are known the angle of advance is fixed; and that **THE ECCENTRIC MUST ALWAYS LEAD THE CRANK AT AN ANGLE OF 90 DEGREES PLUS THE ANGLE OF ADVANCE** (for external valves.)

> **Q1.** Given, throw 2 inches, steam lap 1 inch, and a 33 deg. Required, (a) amount of lead (b) angle of lead, (c) angle of lap, and (d) angle between the crank and the eccentric.
>
> **Q2.** Given, throw 2¼ inches, a 30 deg., and lead ⅛ inch and 3-16″ respectively at H. E. and C. E. Required, the laps.
>
> **Q3.** If there is neither lap nor lead, what is a equal to and what are the crank positions for the opening and closing of the valve?

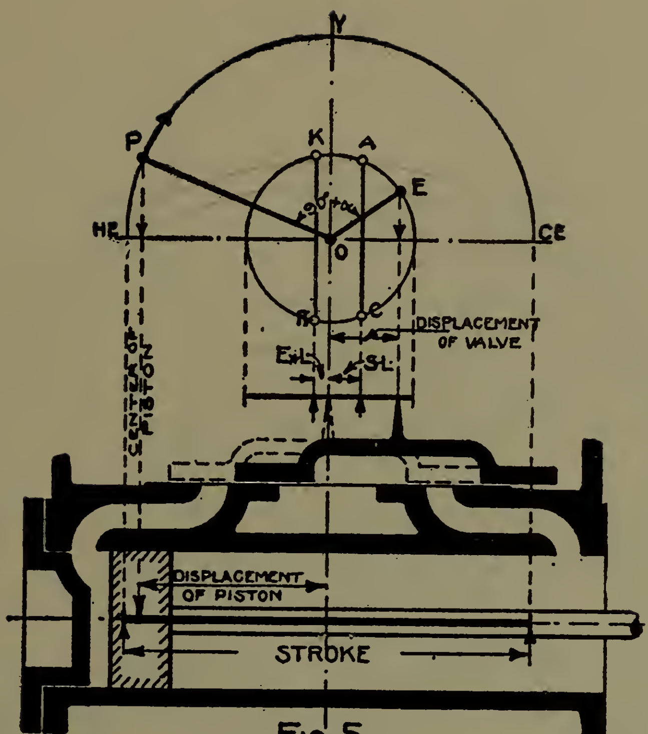
The **four periods** of operation of the valve are admission, expansion, exhaust and compression.

The four **principal valve events** are admission (A), cutoff (C), release (R) and compression (K). The four **minor events** are maximum displacement of the valve to the right (M), same to the left (M′), valve central and moving to the left (Q), and central but moving to the right (q).

The letters given in the parentheses in the above list with be used to indicate the respective events on the diagrams which are to follow. For the principal events of the head end of the valve, capital letters will be used; for those of the crank end of the valve, small letters will be employed.

It will be noticed that each working edge of the valve operates two events, opening and closing. The steam edge operates both adm. and C. O.; the exhaust edge rel. and comp. These pairs of events will be termed **conjugate events.**

If the proportions of the different parts of the gear are known the action of the valve may be studied by drawing the whole mechanism in its different phases. If the valve gear has the slotted crosshead the construction may be simplified by considering the vertical center line of the cylinder to be moved over to coincide with the vertical center line of the shaft, as in Fig. 5, in which case the positions of the centers of the valve and piston may be found by simple vertical projection from the centers of the eccentric and crank pin, in the manner shown in the figure.



FIG·5

Referring to Fig. 5 and considering only the head end of the valve,—when the steam edge is just even with the port edge the eccentric, having a displacement S. L., will be at either A or C (which are vertically above the center of the valve when it is in this position); and when the exhaust edge of the valve is just "line and line" with the edge of the port, the eccentric will be at either R or K. Then, considering the direction of rotation, it is evident that when the eccentric is at A the steam edge of the valve opens, and when at C this edge closes; at R, the exhaust edge opens and at K it closes. The reference letters on the figure, therefore, properly indicate the events according to the system of notation which has been adopted.

In a similar manner the position of the eccentric for the events of the crank end of the valve may be found.

If the horizontal diameter of the crank circle is taken to represent the stroke of the piston, the position of the latter in its stroke may be found by projecting vertically on to that line from the crank pin in the manner shown in the figure. In a similar way the position of the valve in its travel may be found by projecting from the eccentric onto the horizontal diameter of the eccentric circle.

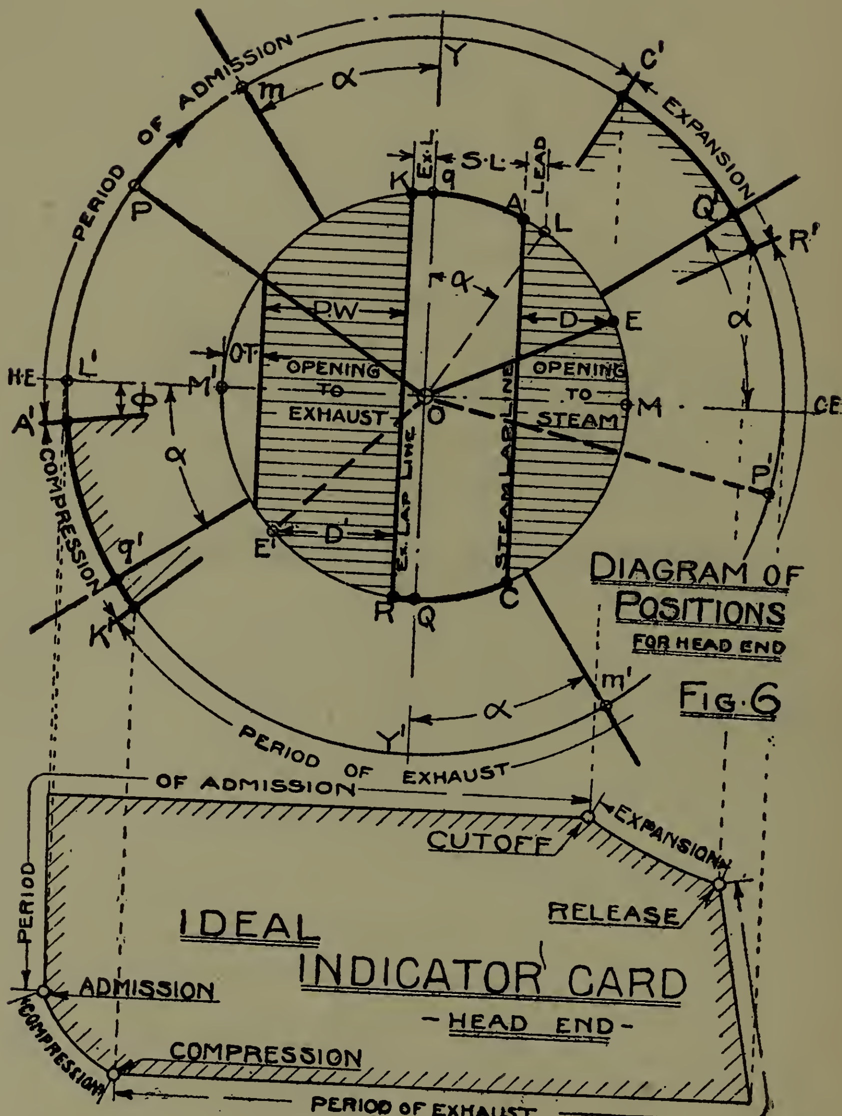

PERIOD OF ADMISSION
EXPANSION
DIAGRAM OF POSITIONS
FOR HEAD END
OPENING TO EXHAUST
OPENING TO STEAM
EX. LAP LINE
STEAM LAP LINE
COMPRESSION
PERIOD OF EXHAUST
OF ADMISSION
CUTOFF
EXPANSION
RELEASE
IDEAL
INDICATOR CARD
-HEAD END-
PERIOD
ADMISSION
COMPRESSION
COMPRESSION
PERIOD OF EXHAUST

FIG. 6

FIG. 7

5. After one has become so familiar with the mechanism as to be able to form a mental picture of the positions of the valve and piston in any phase, the drawing of these may be dispensed with; then only the diagram in the upper part of Fig. 5 need be constructed. This diagram, which will be called the **DIAGRAM OF POSITIONS**, is shown more complete in Fig. 6, for the head end of the valve.

(a) Referring to Fig. 6, as the eccentric rotates and passes the positions which are lettered, the corresponding events or positions of the valve are as follows:

Eccentric at	Valve Event or Position
q	Central,—moving to the right.
A	Admission,—just opening to steam.
L	Lead opening.
M	Extreme opening to steam.
C	Cutoff,—closing to steam.
Q	Central,—moving to the left.
R	Release,—opening to exhaust.
M′	Extreme opening to exhaust.
K	Compression—exhaust closure.

The positions of the crank pin corresponding to those of the eccentric are indicated in Fig. 6 by using the same letters primed, with the exception of m and m′ which correspond to M and M′.

(b) In Fig. 6, the line A C, which will be called the **Steam lap line** (for the H. E.), is to the right of YY′ a distance equal to the steam lap (H. E.). The width of the opening of the valve to admit steam is equal to the distance that the eccentric is to the right of this steam lap line, as shown by D, when the eccentric is at E.

The **exhaust lap line** R K (for the H. E.) is to the left of YY′ a distance equal to the positive exhaust lap (H. E.). If the exhaust lap is negative R K will be to the right of YY′. The width of opening of the exhaust edge of the valve is equal to the distance the eccentric is to the left of the exhaust lap line.

(c) The effective openings of the steam and exhaust edges of the valve are shown by the horizontal sectioning. When the exhaust edge overtravels the edge of the port, the maximum effective opening is equal to the port width (p. w.). The overtravel is O. T. in the figure.

Note the angles through which both the crank and the eccentric rotate during each of the four periods.

The following statements can easily be seen to be true:—

(d) When the **valve is central** the crank is at the angle a behind the dead center position (either H. E. or C. E., as the case may be), i. e. it would have to move through the angle a in the direction of rotation to reach the dead center position.

(e) When the **eccentric is on dead center** the crank is at an angle a behind its vertical position (either up or down).

(f) The **valve is moving to the right** while the crank rotates from m′ to m, i. e. while the crank is to the left of the line mm′.

(g) The valve is **displaced to the right** while the crank rotates 180 degrees, starting at a behind the head end dead center, i. e. while the crank is above the line Q′q′. It is to the left during the rest of the revolution, or while the crank is below Q′q′.

(h) When the steam edge of the valve is just beginning to open the crank is at an angle ϕ (lead angle) behind the dead center position (either H. E. or C. E. as the case may be).

(i) The valve has the same position (displacement) for both the opening and the closing of any one of its edges; but the directions of motion are, of course, opposite.

(j) When the opening of the valve occurs after it, or the eccentric passes the central position, the **lap is positive**; when the reverse is true the lap is negative.

(k) Fig. 7 shows the ideal indicator card corresponding to Fig. 6.

Diagrams simliar to Fig. 6 and 7 can be constructed for the **crank end** of the valve in like manner. When the laps at both ends of the valve are equal, the eccentric and crank positions for the C. E. are diametrically opposite those for the head end of the valve.

X3 Construct a Diagram of Positions for a valve having the dimensions given in X1, the angle of advance being 32½ deg.; (a) for the H. E.; (b) for the C. E.

X4. Construct a Diagram of Positions for a valve having zero laps and no lead; (a) for the H. E.; (b) for the C. E.

X5. Given ϕ equals 5 deg., a equals 30 deg., throw 2 inches, crank radius 3 inches, angle for release 10 deg. Required, both laps, the lead, and the crank and piston positions for all events.

Q4. Considering each of the four valve events in turn, will they be made to occur earlier or later (a) by increasing the laps; (b) by decreasing them; (c) by increasing a; (d) by decreasing a ?

Q5. Are the conjugate events both affected similarly (a) by changing the lap; (b) by changing a ?

Q6. **(a)** Where will the valve be when the crank is at the angle a behind each of the four "quarter-positions" (0 deg., 90 deg., 180 deg., 270 deg.)?

(b) Where will the crank be at the time of admission?

Q7. **(a)** If the outer edge of the valve opens when the crank is 33 deg. behind the dead center and a is 30 deg., is the lap positive or negative?

(b) Same but with a equal to 36 deg.

(c) Same as (a) but for the inner edge.

(d) . . Same as (b) but for the inner edge.

CHAPTER II.

THE ELEMENTARY VALVE DIAGRAMS.

6. VALVE DIAGRAMS. These should show by simple and accurate geometrical constructions the action of all parts of a valve gear throughout the complete cycle of operation. These diagrams are sometimes used for the purpose of **analyzing** the action of valve gears the proportions of which are already known; and at other times for **designing**, that is, to determine the proportions of valves to give certain predetermined steam distributions with certain widths of openings. There are a number of different constructions used for valve diagrams, some of which are better for analysis, and others better for designing. Often it is desirable to use a combination of two or more diagrams for a thorough investigation, as certain features are shown more clearly by some diagrams than by others.

The elements of the more common diagrams will be briefly discussed in this chapter. We will consider only that mechanism which has the slotted crosshead both at the eccentric and at the crank pin.

As none of the valve diagrams show the true positions of the eccentric, it is frequently advisable to show these by drawing little figures opposite the crank pin in its various positions, as is done in several of the illustrations which follow. These little figures will be called **Pilot Diagrams.**

Each of the diagrams which will be discussed should be applied to the solution of the first of the following typical problems and an attempt be made to solve the second. In this way the possibilities and limitations of the various diagrams will become apparent.

These diagrams should be drawn full size, with the exception of the crank pin circle, which may be made any size whatever as we are only interested in the crank angles and in the piston position expressed as a fraction or as per cent of stroke. Often the eccentric circle is used for the crank pin circle. If a circle 6¼ inches in diameter in used then each 1-16 inch is 1 per cent of the stroke.

X6. Analysis. Given the throw 3 inches, angle of advance 31½ deg., both steam laps 1⅜ inches, positive exhaust lap for crank end of the valve ½ inch, negative exhaust lap for head end ⅛ inch and widths of both ports 1¼ inches. Required for both ends of the valve,—

(a) The crank and piston positions for all the events;

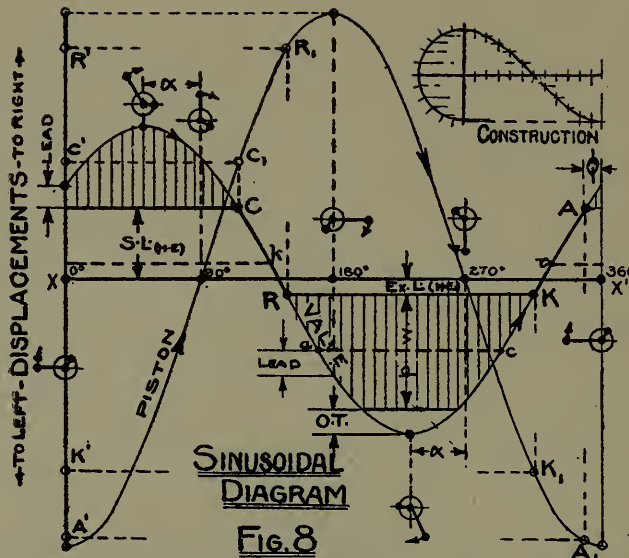
(b) The angles of rotation of crank and eccentric for each of the periods;

(c) The maximum openings to steam and exhaust;

(d) The overtravel (steam and exhaust);

(e) The lead.

X7. Design.—Given cutoff ¾ stroke, (A) the amount of lead ¼ inch or (B) the lead angle 5 deg., the maximum width of opening of the steam edge of the valve 1¼ inch, release 95 per cent of stroke for H. E. and 90 per cent. for C. E. Determine for both ends the value of (a) the angle of advance, (b) throw, (c) steam lap, (d) exhaust lap, (e) crank and piston positions for each event and (f) the overtravel of both the steam and exhaust edges.



SINUSOIDAL DIAGRAM

FIG. 8

7. **THE SINUSOIDAL DIAGRAM.** The harmonic motious of the valves and piston can be shown by using rectangular co-ordinates, plotting the displacements as ordinates on the crank positions as abscissae. The resulting curves are sinuoids. In Fig. 8 the crank angles are measured from the H. E. dead center, the positive ordinates are for displacements to the right, and the negative ones are for displacements to the left.

The method of constructing a sinusoid is shown in the upper right hand corner of the figure. The valve sinusoid crosses the X-axis when the crank angles are (180 degrees minus *a*), and (360 degrees minus *a*), and its ordinates are maximum when the angles are (90 degrees minus *a*), and (270 degrees minus *a*), since these are the respective angles for zero and maximum displacements of the valve.

It will be remembered that, when the displacement is equal to the lap, the valve is either just closing or just opening and that any greater displacement is equal to the width of opening. If, then, "lap lines" are drawn parallel to the X-axis and at distances from it equal to the laps, the intersections of these lines with the valve sinusoids will determine the crank angles for various events, and the portions of the ordinates beyond these lines will represent the valve openings. In the figure the steam and exhaust lap lines are AC and RK for the H.E.; and ac and rk for the C. E. The periods of opening for the head end are shown by the areas which are section lined. The maximum effective opening of the exhaust edge is equal to the port width (p. w.). The overtravel is O. T. The lead openings occur, of course, when the crank angle is 0 deg. and 180 deg.

This diagram requires considerable time and care to construct, especially when accuracy is essential. It is useful for analyzing the action of a valve gear of which proportions are already known, as in **X6**, but can not be used for the solution of design problems like **X7**.

8. THE ZEUNER DIAGRAM. If, using the polar co-ordinates, we lay off the valve displacements (regardless of the sign) as radii-vectores on the corresponding positions of the eccentric arm, as in Fig. 9, the locus of the points plotted will be the two circles with centers at o and o′ and having diameters equal to the throw of the eccentric. These loci will be called **displacement circles,** and in the figure that for displacements to the right is shown by the bold line and that for displacements to the left by fine line. When the eccentric is at E the displacement of the valve is D, and it is to the left.

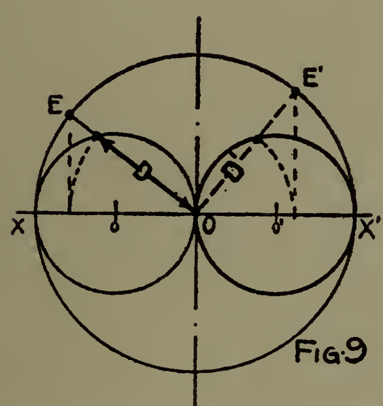


FIG. 9

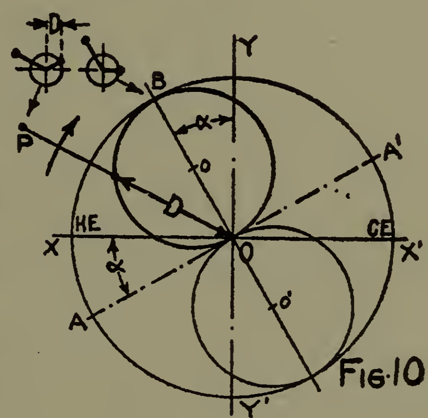
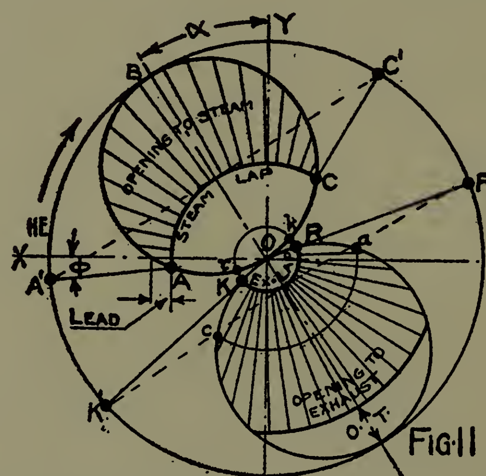


FIG. 10

If the valve displacements are laid off radially along the corresponding **crank** positions (instead of on the eccentric positions as in Fig. 9), we have the Zeuner diagram which is shown in Fig. 10. In this figure, as before, the displacement circle which is shown by the bold line is for the displacements of the valve to the right. When the crank is in the position OP the displacement of the valve is equal to D, and is to the right.

The axis O B of the "displacement circles" is at the angle a behind the Y-axis, for this is the position of the crank when the valve has the greatest displacement. The valve is central when the crank is in either of the positions OA or OA′, at right angles to the axis OB.

In applying this diagram it is often convenient to let the eccentric circle also represent the crank circle, as is done in Fig. 11. In this figure the positions of the crank pin for the valve events are shown by the usual reference letters with primes. The lap lines are evidently arcs of circles with radii equal to the laps, those shown in the figure by the bold lines AC and RK being respectively the steam and exhaust laps for the head end, and those shown by the fine lines ac and rk being the laps for the



FIG·11

crank end. The effective openings (shown by the radial section lines) and the crank positions for the different events are shown for the head only, those for the crank end being admitted to avoid complicating the diagrem. The lead opening occurs, of course, when the crank is on dead center, the amount and angle of lead for the H. E. being shown in the figure. It will be seen that the chords A′ C′ and R′ K′ are tangent respectively to the steam and exhaust lap arcs.

This diagram is simple to understand, easy to construct, and shows at a glance the action of the valve throughout the whole cycle. The true eccentric positions are not shown and when the laps are small (and those for the exhaust usually are) the crank angles and piston positions for the valve events can not be determined very accurately.

This diagram is very useful for analyzing the action of a valve gear when the proportions, including the throw of the eccentric, are known (as in Problem X6). It can also be used to design a valve to meet a given set of conditions (as in Problem X7), but, especially when the throw of the eccentric is not given, this involves rather complicated constructions, which will be omitted as there is another diagram which is much better to use for this purpose.

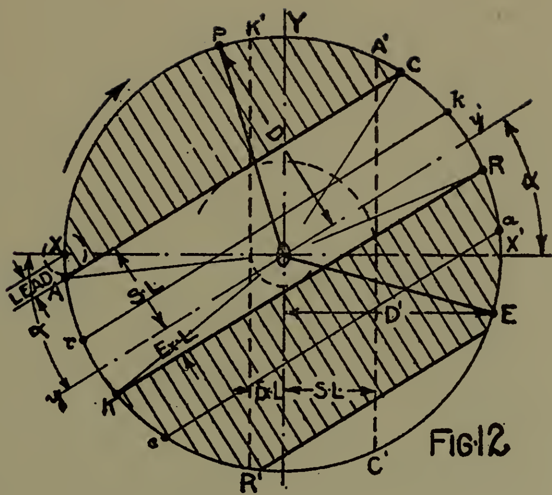


FIG 12

9. SWEET DIAGRAM. In Fig. 12, A′ C′ and R′ K′ are respectively the steam and exhaust lap lines for the H. E. transferred from the Diagram of Positions, Fig. 6. Letting the eccentric circle also represent the crank circle, then, when the eccentric is in positions A′ and C′ the corresponding positions of the crank pin will be A and C. In the Sweet diagrams the chord AC is the steam lap line (H. E.). Similarly when the eccentric pin are R and K and the exhaust lap line (H. E.) is the chord RK. The axis Oy is parallel to the lap lines.

It is evident that if the axis OY and the lap lines of the Diagram of Positions are rotated back through the angle 90 deg. plus a, they will coincide with the similar lines on this new diagram. In the former diagram the lap lines are chords between eccentric positions, in this latter one they are chords between corresponding crank pin positions (taken on the eccentric circle).

It will be evident that the lap lines are at a distance from O equal to their respective laps; that they and the axis Oy are inclined at an angle a with respect to the horizontal axis; that corresponding to any crank position, the displacement of the valve is equal to the perpendicular distance from the crank pin to the axis Oy, (as shown by D when the crank is at OP); that, similarly, the opening of the valve is equal to the perpendicular distance from the crank pin to the lap line; and that the lead, which is the opening when the crank is on the dead center, is equal to the radius of a circle which is tangent to the steam lap line and has center at X (for H. E.).

In the figure the effective openings of the valve (shown by the section lines) and the positions of the crank pin for the various events are shown for the head end of the valve only. ac and rk are the lap lines for the crank end.

In the Zuener Diagram, Fig 11, compare the chords A′C′ and R′K′ with the lap lines on the Sweet Diagram.

This Sweet Diagram gives accurate results, is easy to understand can be applied readily to problems of analysis (Prob. X6) but problems of design (Prob. X7) can be solved only by using more or less complicated constructions, which we need not consider. This diagram is also known as the Reuleaux Diagram.

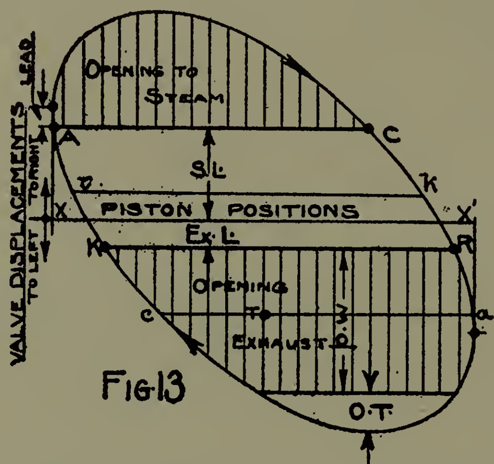


FIG 13

10. ELLIPTICAL DIAGRAM. To show at a glance the simultaneous displacements of the valve and piston throughout the complete revolution of the engine, the displacements of the valve may be plotted as ordinates on the corresponding positions of the piston as abscissae. The resulting figure will be an ellipse as is shown in Fig. 13. To have the generating point move around the ellipse in the direction in which the crank rotates (as shown by the arrows), the displacements of the valve to the right are plotted as positive ordinates (up), and those to the left as negative ordinates (down).

In this diagram the lap lines are AC and RK, for the head end of the valve. The effective openings to steam and exhaust, the points for the various events and the lead are shown in the usual manner for the H. E. For the crank end of the valve the same ellipse would be used but the laps would, of course, be laid off opposite to those for the H.E., as shown by the fine lines ac and rk.

In constructing this diagram the displacements of the valve and of the piston can be obtained from any of the other diagrams, including the Bilgram which is to follow, but most readily from the Zeuner.

This diagram can not be used for the solution of problems in design such at Problem X7, but is of the greatest value for analyzing (Prob. X6) as it shows at a glance the operation of the valve throughout the whole cycle.

11. THE BILGRAM DIAGRAM. The diagrams which have already been discussed were seen to have their limitations. All can be used for analyzing; some are applicable to problems in design only by employing rather complicated constructions, and others not at all; some give accurate results and others do not.

The Bilgram Diagram, which will be considered next, is unlimited in its application, is simple to construct and always gives accurate results. It is especially valuable for designing and is the diagram which we will use most frequently for that purpose hereafter. Unfortunately it is a little harder to understand and does not show the action of the valve throughout the complete cycle quite as clearly as do the Zeuner and Elliptical Diagrams, so these latter and other diagrams will be used to supplement it at times. Like all of the other diagrams, the Bilgram does not show the true positions of the eccentric, since it uses only the crank positions.

(a) In the Bilgram Diagram for the H. E. the angle of advance is laid off above OX′ (Fig. 14) locating on the eccentric circle the point Q, which is called the **lap circle center.** Q is a stationary point and must not be confused with the eccentric center.

Fig. 14

Before showing the manner in which this point is used it is necessary to fix in mind certain of the relative movements and positions of the crank and valve.

In this figure, MM′ is at right angles to Qq and consequently each of the points Q, q, M and M′ is at the angle a behind the nearest axis.

Noting the positions of the crank and eccentric in the little Pilot Diagrams the following statements (b, c, d and e) will be seen to be true.

(b) The **valve is central** when the crank coincides with either OQ or Oq. The valve is **moving to the left** when the crank is at OQ, and to the right when at Oq.

(c) The **valve is displaced to the right** while the crank rotates from Oq to OQ, i. e. while the crank is above Qq. The displacement is to the left when the crank is below Qq.

(d) The **maximum displacement to the right** occurs when the crank is at OM. That to the left occurs when the crank is at OM . The valve will have its **maximum opening or closure** when the crank is in one or the other of these positions, depending on which of the edges of the valve is under consideration.

(e) **The valve is moving to the right** while the crank is rotating from OM′ to OM i. e. while the crank is to the left of MM′. The motion is to the left when the crank is to right of MM′

(e′) Considering the head end of the valve and referring to Fig. 6, it is evident that:—

> The **steam edge closes** when the crank is near OQ.
> The **steam edge opens** when the crank is near Oq.
> The **exhaust edge closes** when the crank is near Oq.
> The **exhaust edge opens** when the crank is near OQ.

> **Q8.** When the valve is moving to the left and is approaching its central position, where will the crank be? When moving to the right and approaching the central position, where will the crank be? Devise other questions of similar nature.

(f) Now, considering the use of the point Q:— In Fig. 15, when the crank OP is on the H. E. dead center the eccentric is at E at the angle α with OY. Now if the crank rotates through any angle β to OP′ the eccentric will of course rotate through the same angle to E′. The eccentric will then be at the angle α plus β with OY and will have a displacement equal to E′D′. From Q draw the line QD perpendicular to the crank OP (produced if necessary). Then it can be seen that the right triangles OQD and O E′D′ are equal. It follows that QD and E′D′ are equal, since they are homologous sides of equal triangles. This is evidently true for all positions of the crank. But QD is the perpendicular from Q to the crank and E′D′ is the displacement of the valve. Therefore, **CORRESPONDING TO ANY POSITION OF THE CRANK, THE DISPLACEMENT OF THE VALVE IS EQUAL TO THE LENGTH OF THE PERPENDICULAR DROPPED FROM THE FIXED POINT Q TO THAT CRANK POSITION** (produced if necessary). This is the **FUNDAMENTAL PRINCIPLE OF THE BILGRAM DIAGRAM.** To get the displacement of the valve corresponding to a given crank position, it is only necessary to measure the perpendicular distance from the fixed point Q to the crank. The position of the eccentric need not be found. If one constantly remembers this Fundamental Principle and will picture in his mind the positions and motion of the valve corresponding to the various positions of the crank (as explained in **b, c, d** and **e**) there will be little difficulty

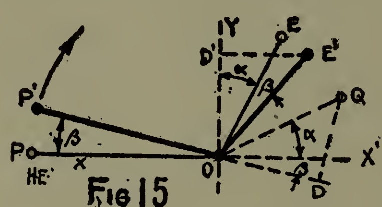


Fig 15

in using the Bilgram Diagram. If one has difficulty in forming these mental pictures, he should draw the little Pilot Diagrams opposite the crank in its different positions.

(g) There will be considerable use for the term **"perpendicular"** in connection with the application of this diagram, so to avoid the necessity of explaining the manner in which the term is employed each time, it will always be understood to refer to the perpendicular from Q to the crank and may mean either the perpendicular direction or the length of the perpendicular, as the context will indicate.

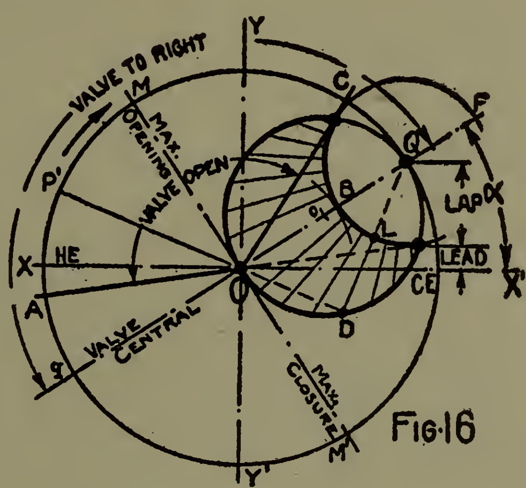


Fig. 16

(h) The **locus of D the foot of the perpendicular** is evidently a circle with OQ as diameter, (why?) as is shown in Fig. 16 for the head end of the valve.

(i) By subtracting the lap, QL (same fig.) from the valve displacement QD, the width of valve opening LD is found. Since the lap is constant, a **"lap circle"** may be drawn with Q as center and with radius equal to the lap, then the **valve openings** are the portions of the perpendiculars outside of this circle, (as shown by the shaded area in the figure).

(j) The **width of valve opening is zero** when either crank itself, or its extension, is tangent to the lap circle; one position of the crank being for the opening of the valve, and the other for the closing, depending on which way the valve is moving and which edge is operating. The foot of the perpendicular and point of tangency coincide when the crank is in either of these positions.

(k) The **closing of the outer edge** of the valve occurs when the crank itself is tangent to the lap circle, (no matter on which side). For, considering the head end of the valve, when the

outer edge is closing the crank will be near Q (see e′) and consequently the crank itself will be tangent.

(l) **The closing of the inner edge** of the valve occurs when the extension of the crank is tangent to the lap circle. The **opening** of this edge of the valve occurs when the crank is tangent to the lap circle on the side opposite that to which it is tangent for closing.

> **X8.** On **X3** draw the steam and exhaust lap circles for a Bilgram Diagram and see if the crank positions are as stated above.

(m) The **lap is positive** if the closing occurs when the point of tangency is on the "back side" of the lap circle, for the closure will then take place before the crank reaches Q, i.e, before the valve has reached its central position.

(n) The **lap is negative** if the closing occurs when the point of tangency is on the forward side of the lap circle. When the lap is negative use broken lines for the lap circle.

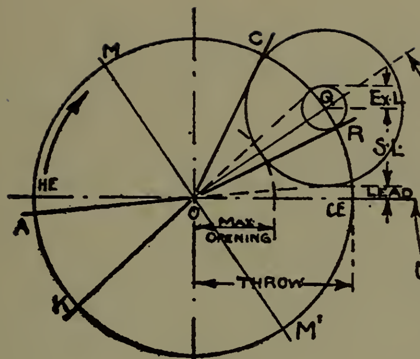
> **Q9.** Considering the H. E. of a D-valve, (a) what are the directions of displacement and of motion when the steam edge with positive lap is just opening? (b) Then on which side of the lap circle will the tangency be, and will the crank or its extension be tangent to the circle? Devise similar questions for the other edge for both opening and closing and both positive and negative laps. Answer by reasoning and not by applying rules.

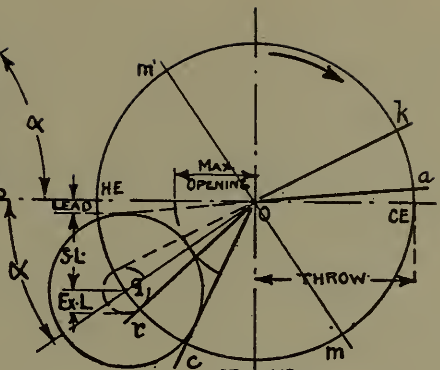
(o) The **lead opening** of the valve is equal to the shortest distance from the lap circle to the X-axis, since this is the opening when the crank is on the dead center. A line drawn parallel to OX′ and at a distance equal to the lead opening above it, will be tangent to the lap circle.

(p) The **maximum opening** of the valve is equal to OB (Fig. 16). Then if with O as center and with radius equal to the maximum width of valve opening, an arc is struck, it will be tangent to the lap circle at B. The **maximum closure** is equal to OF.

Fig. 16 is for the H. E. outer edge of the valve, the lap being positive. The valve is open when the crank rotates from A to C and is closed during the remainder of the revolution. It is evident that OQ bisects the angle between OC and the extension of OA.

(q) In a **Bilgram Diagram for the crank** end of the valve the lap circles would be drawn with centers at q, diametrically opposite Q, and all the preceeding statements are still applicable.



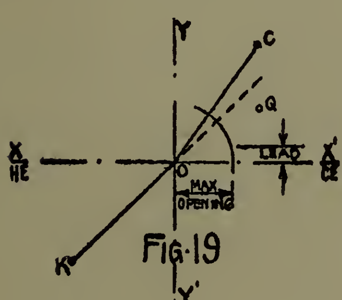


Fig·17 Fig·18

(r) Complete Bilgram Diagrams are shown in Fig. 17 for the head end of a valve and in Fig. 18 for the crank end. The angle of advance and throw of the eccentric must be the same in both these diagrams since they refer to the same eccentric, but the laps of the two ends of the valve are not the same. The exhaust lap for the crank end is negative and is therefore shown by a dotted circle. These two diagrams could be combined into one, but are shown separated here to avoid confusion.

(s) The **application** of the Bilgram Diagram to problems in **analysis,** like X6, is very simple. After the eccentric circle has been drawn, the lap circle center Q is located and the lap circles for steam and exhaust edges are drawn. The crank positions for the various events are tangent to the lap circles.

(t) The **application to design** problems when the cutoff, lead opening and maximum valve opening to steam are known, is as follows:—In Fig. 19 for the H. E. of the valve, starting with the X and Y axes, draw the crank position OC for cutoff; draw a line parallel to OX and above at a distance equal to the lead; and with O as center and radius equal to the desired maximum valve opening strike an arc in the position shown. From what has gone before it is evident that the steam lap circle must be tangent to these three lines. The location of its center Q can usually be found as quickly and as accurately by trial, as by geometrical construction. Having the point Q determined, the throw and angle of advance of the eccentric and the lap are known.



Fig·19

If, in Fig. 16, instead of the lead opening, the lead angle ϕ is known, Q will be so located that the lap circle will be tangent to OC, to OA produced and to the arc for width of valve opening.

If now, the crank position OK for compression is drawn (Fig. 19), the exhaust lap circle will of course have its center at Q and will be tangent to this line produced. The crank position for release will be tangent to this same circle on the other side.

12. PSEUDO-POLAR DIAGRAMS is the term which will be used as referring to the Zeuner, the Sweet, and the Bilgram Diagrams.

CHAPTER III.

ARRANGEMENTS AND LIMITATIONS OF THE SIMPLE VALVE GEAR. ANGULARITY OF RODS AND EQUALIZATION OF VALVE EVENTS.

ARRANGEMENTS OF THE SIMPLE VALVE GEAR.

13. "RUNNING UNDER" or backward rotation. Applying the general rule for external valves, that the eccentric must lead the crank at an angle of 90 deg. plus a , the position of the eccentric with respect to the crank for back ward rotation is that shown in Fig. 20.

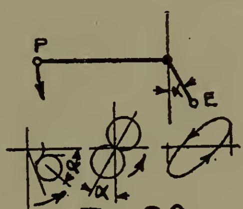


FIG. 20

Q10. In turning the eccentric to reverse an engine what will be the angle between the new keyway and the old?

It is evident that the valve diagrams would be reversed with respect to the horizontal center line, from those for engines which "run over," as is shown in the lower part of the figure for the Bilgram, Zeuner and Elliptical Diagrams.

X9. Same as X6 but for engine running under.

14. INTERNAL VALVES. Since the motion and displacements of the internal valve must be just opposite from those of the external valve, the **eccentric must be diamétrically opposite** that for the latter case and therefore **will follow** the crank at an angle of 90 deg. minus a , as in Fig. 21.

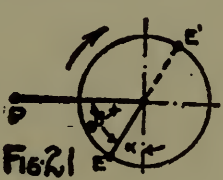


FIG. 21

Q11. Where will the eccentric be when an internal valve is used on an engine which runs under?

To avoid confusion, in constructing the valve diagrams for internal valves it is advisable to ignore the fact that the valve is in internal, and draw the diagrams the same as for external valves.

15. VALVE ROD GUIDES. It is usually advisable to guide the end of the valve rod in some manner and this may be done by using either a sliding guide (crosshead), or a rocker arm. If the **rocker arm** is used the pin moves in a circular arc, and consequently has movement sidewise as well as longitudinally. (See

Fig. 22). It is necessary to make provision for this lateral movement of the end of valve rod and this may be done either by using a rod which has considerable flexibility or by introducing in the rod some form of "knuckle joint." The arc should be so located that the lateral movement of the pin is the same in amount on either side of the center line of the rod. Of course the longer the arm of the rocker, the less will be this side movement.

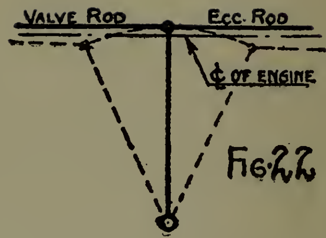


FIG. 22

When the end of the eccertric rod is joined to the end of the valve rod in such a manner that the valve and eccentric nominally have the same amounts and directions of displacement, the arrangement will be called a **"direct drive."**

If the guiding rocker is pivotted half way between the pins for the eccentric rod and valve stem, as in Fig. 23, the motion of the valve is of course reversed from that derived from a direct drive, so either the eccentric must be placed diametrically opposite that for the latter case or else an internal valve must be used. This arrangement having the **reversing rocker** may be conveniently used when the center lines of the valve rod and eccentric rod are some distance apart.

FIG. 23

Q12. On an engine which runs under and has an internal valve and reversing rocker, where would the eccentric be placed?

Sometimes the rockers are arranged to give the valve either a greater or a smaller movement than that derived directly from the eccentric. These are termed **multiplying** or **reducing** rockers as the case may be and are shown in Fig. 24.

16. In constructing the valve diagrams when this last type of rocker is employed, it is convenient to use an imaginary, or **"Virtual" Eccentric**, which is of such size that it would cause the valve to have the same motion if a direct drive were used. The diameter of the virtual eccentric circle is therefore equal to the travel of the valve.

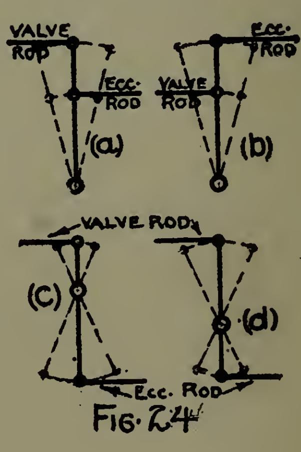


FIG. 24

The diagrams will show the true values for the laps, the openings of the valve, and the angle of advance, and will give the true angular positions of eccentric and crank; but the throw is, of course, not that of the actual eccentric. A drawing showing the true arrangement of the linkage should accompany the valve diagram.

> **Q13.** What is the radius of the virtual eccentric when a multiplying rocker having a ratio of arms 3 to 2, is used with an eccentric of throw equal to 2 inches?

17. DISTORTION OF VALVE MOTION BY ROCKER ARM. The travel of the valve is of course equal to the perpendicular projection of the arc of the rocker pin onto the center line of the valve stem, and the valve is central with its seat when the pin is in the middle of its arc. In Fig. 25, in which the arc is oblique to the center line of the valve stem, the valve is central when the pin is at P. It is seen that, with this arrangement of rocker, the motions of the valve to either side of its central positions are not the same, as shown by the difference between a and b. In order that these motions shall be equal, it is evident that the rocker arm must be at right angles to the valve stem when the valve is cen-to the valve stem when the valve is central. It will also be apparent that the rocker arm must at the same time bear the same relationship to the eccentric rod. To cover the case of rockers having separate arms for the pins of the valve rod and eccentric rod, the general statement may be made that **the center lines of the arms of the rocker must be at right angles to their respective rods when the valve is central, to avoid distorting the valve motion.**

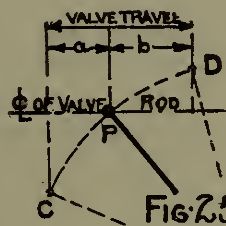


FIG. 25

If the eccentric and valve rods are not parallel a **bent rocker arm** or **bell crank** must be used, as in Fig. 26 b, to satisfy the rule just given.

> **Q14.** What is the relation between the angle between the rocker arms and that between the rods?
>
> **Q15.** Sketch several arrangements of both the direct and reversing types, taking the rods at different angles.

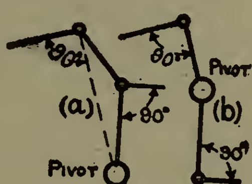


FIG. 26

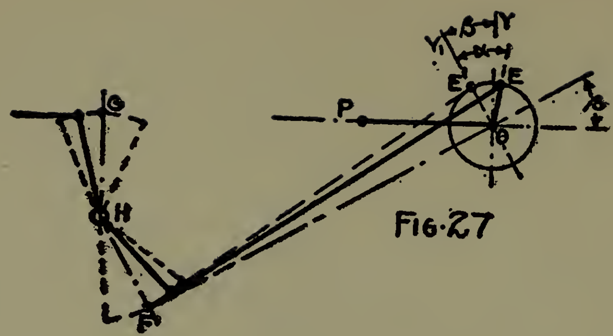


Fig. 27

18. POSITION OF ECCENTRIC WITH OBLIQUE ECCENTRIC ROD. Referring to Fig. 27, when the valve is central the rocker arm will be in the position FHG, and the eccentric will be nominally at E′ at right angles to the line OF, which is the mean or "nominal" position of the eccentric rod. When the crank is on the dead center P, the eccentric must evidently be advanced the angle a ahead of this position E′, and will consequently be at E. No matter what sort of guide is used for the end of the eccentric rod it is evident that the **general rule for the position of the eccentric** is as follows for external valves:—When the crank is on the H. E. dead center the eccentric will be (90 deg. plus a) ahead of the nominal position of the eccentric rod. Only when the eccentric rod is parallel with the center line of the engine will the eccentric be (90 deg. plus a) "ahead of the crank," as the rule is usually stated. Fig. 27 is for an internal valve.

Q16. What is the angle between the crank and the eccentric when the eccentric rod is inclined upward at 30 deg. and the angle of advance is 30 deg. (a) for external valve with reversing rocker, (b) for external valve with direct drive, (c) and (d) same as (a) and (b) but for internal valve.

Q17. On an engine having two cylinders and two cranks at 90 deg. how may the same eccentric be used to drive both valve gears? (b) Same, when the angle between the cranks is 120 deg.

ANGULARITY OF THE CONNECTING ROD.

19. DISTORTION DUE TO ANGULARITY. In Fig. 28, o is the middle of the stroke and the distance oO is equal to the

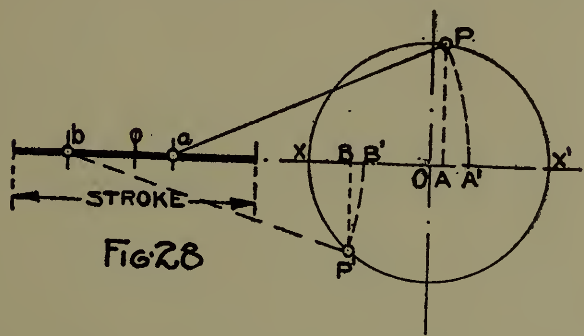


Fig. 28

length of the rod aP. If an infinite rod is used, the displacement of the piston oa will of course be equal to the displacement of the pin P which is equal to OA. If, however, a finite rod is used these displacements will not be equal. For, if the end of the rod a is kept stationary and the other end P is uncoupled and swung to A′, then oa will be equal to OA′, which is seen to be greater that OA. It will be found that no matter where the crank is A′ will always be to the right of A. It is evident that owing to the **angularity** of the connecting rod, if one of finite length is used, **the piston is always nearer the crank end of the stroke** than it would be nominally, except of course when it is at the end of its stroke.

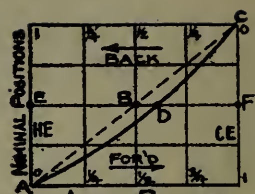
It follows that:— **The valve events occur later in the forward stroke and earlier in the return stroke** than they would nominally.

The distance AA′ is the "distortion due to the angularity of the rod" and is equal to the difference between the length of the rod and its horizontal projection. This distortion is greatest when the crank is at right angles to the center line of the engine and decreases to zero at the ends of the stroke. The shorter the length of the rod when compared to the crank radius, the greater is this relative distortion.

If the diameter of the crank circle XX′ represents the stroke of the piston, then, having any position, such as A′, the corresponding position of the crank pin P may of course be found by drawing the "connecting rod arc" A′P; or if P is known at the start, A′ may be found from it in a similar manner.

The way the distortion varies throughout the stroke is shown by Fig. 29, which may be called a **Diagram of Distortions.** Plot-

ting nominal piston positions as ordinates and true piston positions as absciassae gives the curve ADC. When the piston has traveled nominally half the stroke, if the movement is from the H. E., it has actually traversed the distance ED; and if the piston is in the return stroke, its distance from the crank end is FD, which is much less, than ED. The scale for the forward stroke is at the bottom of the diagram and that for the back stroke is at the top.



ACTUAL POSITIONS.
FIG. 29

If, now, the nominal displacements are plotted as abscissae on the same ordinates as before the straight line ABD is of course obtained. Then the horizontal distances between this line and the curve ADC are the distortions. When the piston has traveled nominally half stroke the distortion is BD. If the movement is from the H. E. the piston will be this distance beyond half stroke and if from C. E. it will be the same distance short of mid-travel. It is evident that the difference between the actual distances traveled from the two ends of the stroke is twice the distortion, and that the mean between these distances is the nominal distance traveled.

On "high speed" engines the length of the connecting rod is usually six times the crank radius; on "low speed" engines it is often 5½ cranks; and on marine engines it is sometimes even as short as 4 cranks.

X10. Draw a Diagram or Distortions to any suitable scale, the length of the connecting rod being six times the crank radius. Tabulate the per cent. difference between the events at the two ends of the valve, the events occurring nominally as follows:— c.o. ¾ stroke, release .90; comp. .85. Same for connecting rod lengths equal to 5, 5½ and 6½ cranks.

20. VALVE DIAGRAMS CONSIDERING "ANGULARITY." All the pseudo-polar valve diagrams show the true positions of the crank. Therefore if the positions of the piston are not being considered, but only those of the crank, the angularity of the connecting rod would not affect the diagram. If, however, after the crank positions have been found, the true positions of the piston are desired, it will then be necessary to consider the angu-

larity. Having already determined the crank positions, the corresponding true position of the piston would be found by drawing the connecting rod arcs in the manner shown in Fig. 30a and b for the Bilgram and Zeuner Diagrams. Should the piston positions be known at the outset, then by drawing similar arcs, the true crank positions can be found and these would be used in constructing the rest of the diagram.

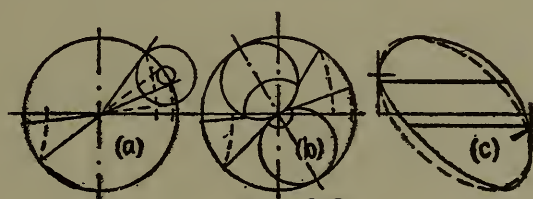


FIG. 30

In the Elliptical Diagram, Fig. 30c, it is evident that the angularity causes all points on the ellipse to be displaced towards the crank end of the stroke. The resulting figure is of oval shape, in consequence of which the diagram is sometimes called the Oval Diagram. The Sinusoidal diagram is distorted in a similar manner.

X11. In X6 find the true positions of the piston for each of the events on each of the diagrams, when the connecting rod is six times as long as the crank.

X12. Using the data of X7 construct a Bilgram diagram considering the angularity of the connecting rod of "length equal to six cranks."

21. THE MULLER CIRCLES. In Fig. 31, O is the center of the shaft, O P′ is the crank, a′P′ is the connecting rod, OA is the frame, AB the stroke, θ the angle between the crank and the frame, and a′ A is the distance the piston is from the head end of the stroke. When the crank pin is at P, on the dead center, the end of the connecting rod coincides with A. Keeping the crank stationary in the position OP and rotating the rest of the mechanism about it, the frame will turn about O, the points A and B describing the circles 3 and 1, and the connecting rod will

rotate about P, its end describing the circle 2. 1, 2 and 3 are the Muller Circles.

When the frame has rotated through the angle θ to the position O A′ the end of the connecting rod is at a on circle 2, and A′ a, which is equal to A a′, is the distance the piston is from the head end of the stroke, properly corrected for the angularity of the connecting rod. Since O A′ coincides with O P′ produced, it is evident that, when the frame is in its normal position OA, to find the true location of the piston in its stroke it is only necessary to prolong the crank to intersect the Muller Circles.

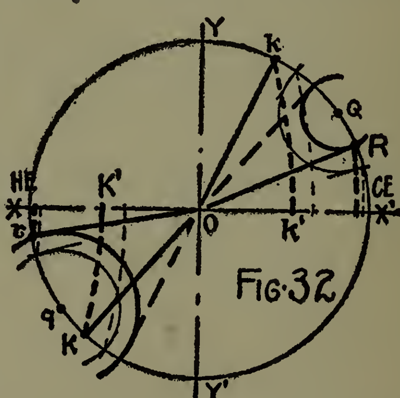
These Muller Circles may be drawn concentric with any ot the pseudo-polar valve diagrams.

X13. Around the diagram X12 draw the Muller circles and compare the piston displacements obtained therefrom for the different valve events with those used in constructing the diagram.

EQUALIZATION OF VALVE EVENTS.

When a finite connecting rod is used with a valve having the same amount of lap at both ends, the displacements of the piston for similar events in the two strokes are unequal, or, as it is usually stated, the valve events are unequal. The expedients which may be employed with more of less success to equalize the events are, (1) to use unequal laps, (2) to distort the valve motion by using an oblique guide for the end of the eccentric rod, and (3) to use a combination of (1) and (2).

22. EQUALIZATION BY USING UNEQUAL LAPS. In the Bilgram Diagram in Fig. 32, in which Q and q are respectively the centers of the head and crank end lap circles, let it be desired to have the exhaust edge of the valve close when the piston is at k′ in the forward stroke and at K′ on the return, these points being at equal distances from their respective ends of the stroke. Striking the connecting rod arcs in the usual manner, the corresponding true crank positions Ok and OK are found. For the compression to begin when the crank is in either of these positions, the back side of the proper lap circle must be drawn tangent to the extension of the crank, as is shown in the figure by the bold lines. The lap circles for the two ends of the valve will be unequal, that for the head end being the smaller (when positive). Using these



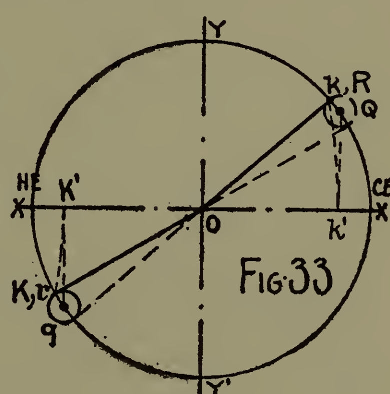
FIG·32

unequal laps, this one valve-event will be equalized. But each of these edges of the valve operates another event (opening or release) which occurs when the crank itself is tangent to the lap circle on its front side, the crank being at OR and Or. It will be found that this event will not be equalized by using these lap circles but that the inequality will be less than that which occurs when the laps are equal.

For the sake of comparison the "nominal" lap circles (which are a mean between those just found for the two ends of the valve), together with the corresponding true positions of the crank and piston, are shown in fine lines in this same figure.

In the foregoing case the nominal lap was assumed to be quite large.

In the special case when there is **zero nominal lap**, as in the Bilgram Diagram shown in Fig. 33, K′ and k′ are the perpendicular projections of Q and q on the horizontal axis, and the correction will give a small **negative** lap circle at Q (H. E.) and a (practically) **equal positive** circle at q (C. E.). Then, the crank position for closure at one end will (practically) coincide with its position for opening at the other end, i. e. K falls on r and k on R. Therefore, since the closure has been equalized, the opening will also be.



Fig·33

In the foregoing cases only the exhaust lap has been considered, and for this it is seen that the head end lap is the smaller, algebraically. If the steam events are equalized the crank end lap will be the smaller.

It is seen that, **in general by making the laps unequal one event may be exactly equalized, and the inequality in the conjugate event be reduced; and that in case the nominal lap is zero, the exact equalization of both events is possible.**

It is usually considered to be more important to equalize lead and compression than cutoff and release. For equal leads the steam laps will be equal.

X14. Throw 2 inches, $a = 33$ deg., compression 85 per cent. of stroke. Construct a Bilgram Diagram, (a) for the nominal exhaust laps; (b) for exhaust laps giving equalized compression; (c) tabulate the true piston positions, in per cent. of

stroke, for both compression and release, for both (a) and (b).

X15. Same as the preceding exercise, but for the case of zero nominal laps.

23. EQUALIZATION BY USING OBLIQUE GUIDE. It is assumed that the valve diagram has been constructed (neglecting the angularity of the rods) to determine the angle of advance and the throw of the eccentric.

In Fig. 34, which is a diagram showing the true positions of the various parts of the valve gear, the larger circle is for the crank pin, and the smaller one is for the eccentric. On the diameter of the crank circle, locate the positions of the piston, in both strokes, for equal cutoffs and equal admissions and note that these events are conjugate. By drawing the connecting rod arcs from these positions of the piston, find A′, C′, a′ and c′, the true positions of the crank pin for these events. The corresponding positions of the eccentric, A, C, a and c, are determined next. Now with A and C as centers and with radius equal to the length of the eccentric rod, strike arcs intersecting at B; and in

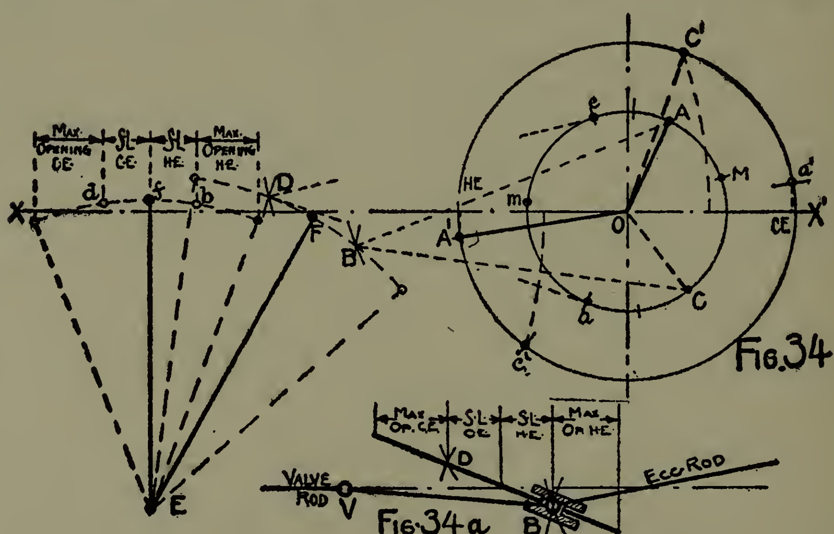


Fig. 34
Fig. 34a

a similar manner, with a and c as centers get the intersecting arcs D. Remembering that the valve, and consequently the guide for the eccentric rod, will occupy the same position for both the opening and closing of any one of the edges, it is evident that if the end of the eccentric rod is guided so as to pass through the points B and D, this pair of conjugate events will be equalized. In the figure an oblique rocker arm FEf, pivoted at E, is employed as a guide, the eccentric rod being attached to pin F and the valve rod to f. The pivot E might have been placed above the axis instead of below it, or a

crosshead with an oblique guide could be substituted for the rocker.

The rocker arm should be so designed that, when the eccentric rod pin is at F, half way between B and D, the other arm of the rocker Ef, will be at right angles to the valve rod. Then the laps at the two ends of the valve will be equal; but they will usually be slightly different in amount from that found by the valve diagram. If b, f and d are the positions of the valve rod pin corresponding to B, F and D, the amount of lap at the head and crank ends of the valve will be the horizontal distances between b and f and between d and f, respectively.

The extreme positions of the rocker pin for the eccentric rod may be found by striking arcs with O as center and with radius, first, equal to the sum of the length of the eccentric rod and the throw of the eccentric, and then, equal to their difference. It will be found that the maximum openings of the two ends of the valve will be unequal and different from those which were used in the valve diagrams; but these differences are usually so slight as to be of no consequence. However, there may be exceptions to this last statement, and in such cases it will be necessary to increase the throw of the eccentric, or to lengthen the rocker arm Ef. If this latter is done, the eccentric rod will be oblique with the center line of the engine, and the angle between the crank and eccentric will be changed to correspond.

In Fig. 34a, the end of the eccentric rod is guided by a crosshead sliding obliquely along BD, and the valve rod is joined by the link VB to the same pin to which the eccentric rod is attached. Neglecting the angularity of the link VB, which is shown abnormally short in the figure, it is evident that the horizontal distance between B and D is the sum of steam laps of the two ends of the valve, and that if these laps are equal the valve will be central when the crosshead pin is at F half way between D and B. The laps and maximum openings are shown in the figure.

Instead of an oblique crosshead a simple rocker arm could be similarly employed.

Usually only one pair of conjugate events can be equalized by this method, for, if besides the intersecting arcs for admission and cutoff, others for release and compression, are drawn, there would then be four points, B, D, H and I, in Fig. 35, through which to pass the arc of the rocker pin and this is usually an impossible construction. It is generally considered to be more important to equalize Admission and Cutoff than release and compression.

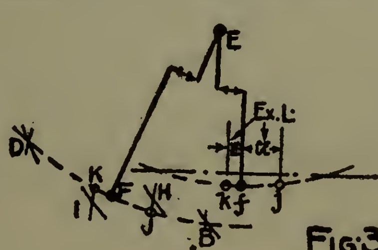


Fig. 35

The **equalization of both release and compression,** in addition to admission and cutoff, may be sometimes approximated closely enough by making the arc of the rocker pin come as near to passing through the intersections H and I (Fig. 35) as is possible, "splitting the difference."

In the special case **when the nominal lap is zero,** the exact equalization of both exhaust events, in addition to admission and cutoff, is possible, for then the intersections H and I will coincide and there will be only three points for the arc to pass through.

X16. (a) Construct a nominal Bilgram Diagram for a valve to have ⅛ inch lead, C. O. ¾ stroke, compression 90 per cent, throw 2 inches. Find the angle of advance and the crank and piston positions (nominal) for admission.

(b) With a length of connecting rod equal to six cranks and using an eccentric rod 15 inches long, equalize admission and cutoff, and approximate the equalization of the exhaust events, by using a rocker similar to that shown in Fig. 34. Take EF 10 inches long.

If the opening of the valve at either end is too small, the arm Ef may be lengthened, but at the same time it must be made perpendicular to an oblique center line passing through O. This new line is then the new center line of the engine, and while it is oblique on the paper it is horizontal on the engine. The angle between this line and the horizontal is the inclination of the eccentric rod on the engine and the position of the ecc. with respect to the crank must be changed to correspond. Instead of lengthening the arm Ef, the throw may be increased. What are the dimensions of the steam and exhaust laps and openings?

X17. Same as preceeding but using a crosshead guide instead of a rocker.

24. OBLIQUE GUIDE AND UNEQUAL LAPS. Having equalized the steam events, **one of the exhaust events** (preferably compression) **may be equalized** by making the exhaust laps unequal. Having located the positions of the eccentric for the equalized compression, from these points as centers and with radius equal to the length of the eccentric rod, arcs are struck intersecting the arc BD, in Fig. 35, at J and K. The corresponding positions of the valve rod pin are j and k, and the exhaust laps will be the horizontal distances between f and j and between f and k. Release will of course not be equalized.

X18. In the two preceeding problems find the exhaust laps for equalized compression and then find the crank and piston positions for release.

ANGULARITY OF THE ECCENTRIC ROD.

25. DISTORTIONS DUE TO ANGULARITY. Just as the angularity of the connecting rod affected the positions of the piston, so also the angularity of the eccentric rod affects the positions of the valve, causing it to be **always nearer the crank end of the stroke** than it would be nominally. This of course affects the valve events, making some to occur later and others earlier.

In the Diagram of Positions shown in Fig. 36, when a Scotch yoke is used in the valve gear, the H. E. and C. E. steam lap lines are respectively AC and ac, and the corresponding leads are the distances ED and ed. Now, employing an eccentric rod in place of the yoke, if, when the steam edge of the H. E. of the valve is even with the edge of the port, the rod is uncoupled at the eccentric and that end is swung across the shaft, it will draw the "steam lap arc" A′ B C′. The distance the eccentric is to the right of this arc is the steam opening for the H. E. The steam lap arc for the C. E. is a′ b c′ and the distance the eccentric is to the left of this line is the steam opening for that end.

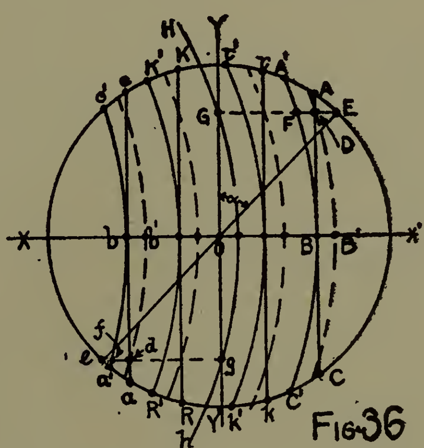


FIG 36

It is seen that the lead opening EF for the H. E. is greater than ED by the amount FD; that for the crank end the opening ef is less than ed by the same amount; and that the events, instead of occurring when the crank pin is in the positions A, C, a and c, will occur when it is at A′, C′, a′ and c′.

26. CORRECTING FOR ANGULARITY OF ECCENTRIC ROD. Now, it is seen that if either the valve rod or the eccentric rod is lengthened an amount equal to FD, the steam lap arcs will be shifted so as to pass through D and d, respectively, thus completely correcting the leads and changing the steam laps to be equal to OB′ and Ob′. Further, the cutoff will be very nearly corrected, as will also be the exhaust events, for the exhaust lap arcs R′ K′ and r′ k′ will be moved over to very nearly pass through the points R and K and r and k. When the valve is central on its seat, the end of the rod would trace the arc Hh,

which crosses the Y-axis at points G and g, which coincide with the horizontal projections of E and e onto that axis.

In Fig. 36 the eccentric rod was purposely taken abnormally short, to cause the distortions to be exaggerated. The longer the rod is, the smaller will be the distortions and with the usual lengths they are so small that the designer may safely neglect them altogether and construct the diagrams the same as for the case of the valve gear having the Scotch yoke. Then, when the valves are "set" in the shop, either the eccentric rod or the valve rod, which are usually adjustable, may be lengthened sufficiently to correct the leads.

> **X19.** On the diagram of **X3** show the correction for equalized lead when the eccentric rod is 30 inches long.

27. THE VALVE DIAGRAMS CONSIDERING ANGULARITY. In case the eccentric rod is unusually short, or for some other reason, it is desired to draw diagrams showing the effect of the angularity of the eccentric rod, the constructions of the various diagrams must be modified. In the Diagrams of Positions it has been seen that the lap lines are eccentric rod arcs. Since the Sweet Diagram is really this former diagram rotated back through the angle 90 degrees plus a, it also would have arcs for steam lap lines.

In the Zeuner Diagram the displacement curves will no longer be circles but will be oval figures, that for displacements to the right being broader, and the one for those to the left being narrower than the circles. These displacement curves may be obtained by plotting as radii vectors the true valve displacements on the corresponding crank angles.

In the Bilgram Diagram, the fundamental principle is not applicable to this case, so the diagram can not be used.

In discussions hereafter the effect of the angularity of the eccentric rod will be ignored.

> **X20.** On the diagrams **X6** show the correction for the angularity of an eccentric rod which is 20 inches long.

28. LIMITATIONS OF THE SIMPLE VALVE. It is impracticable to have cutoff occur much before ⅝ stroke, because, in order to obtain a satisfactory width of opening with earlier cutoff, it is found (1) that the valve and eccentric are excessively large, (2) that consequently the valve gear must work against great friction and inertia forces, and (3) that the release and compression occur too early in the stroke.

> **X21.** Construct a nominal Bilgram Diagram for a valve which is to cutoff at ¼ stroke, have 1 inch maximum opening to steam, and ⅛ inch lead. (a) What are the throw and angle of advance? (b) With release at 90 per cent. of stroke, where will compression begin?

CHAPTER IV.

FACTORS USED IN DESIGNING—AREAS OF PASSAGES, DISTRIBUTION OF STEAM, DRAWING THE VALVE, ETC.

In this chapter the discussion will not be limited to the case of engines having the simple D-valve, but will be extended in some instances to include also those types which are still to be considered. The matter relating to these latter cases may be omitted until the study of them is taken up later.

29. SPEED AND VELOCITY. These terms are often used ambiguously. When the "speed of the engine" is spoken of, the rotative speed of the crank is meant and this is usually expressed in revolutions per minute. By piston speed" (V) or velocity, the mean velocity of the piston is generally meant. This is usually expressed in ft. per minute, and is equal to twice the stroke (L), in feet, multiplied by the r. p. m. (N), i. e. $V=2LN$. The same term may also be applied to the instantaneous velocity of the piston when in some particular position in its stroke. Similarly, the term velocity of steam may mean either the mean or the instantaneous value, but usually the former. The context generally indicates in what sense the terms are used, if they are employed loosely.

30. COMMERCIAL TYPES OF STATIONARY ENGINES. It is unnecessary here to discuss the different types or arrangements of engines except possibly to explain very briefly what is meant by the terms "high," "medium" and "low speed engines" as applied to the better grades of commercial types, as these terms will be used later. By "speed," when used in this connection, is meant the rotative speed. Of course for a given "piston speed" the greater the stroke is, the smaller will be this rotative speed.

HIGH SPEED ENGINES are those which have high rotative speeds accompanied by strokes which are very short when compared to the diameter of the cylinder, the piston speed being generally in the neighborhood of 600 ft. per minute. The stroke is

usually about equal to the diameter of the cylinder, sometimes more and sometimes less. These engines almost always have a single, balanced valve and a shaft governor. They are often called "short stroke engines," and are designed to occupy the smallest space, have the least weight, and "direct connect" to the smallest dynamo for a given power, of any of the stationary commercial types. This class includes only engines of comparatively small power, the cylinders being not usually made larger than 20 inches in diameter.

LOW SPEED ENGINES are those having long strokes (from 2 to 4 times the diameter of the cylinder) and usually operate at less than 120 r. p. m., the speed being limited by the valve gear, the action of which becomes unreliable at higher speeds. This class includes engines having the Corliss and other types of trip cutoff gear. The governor is usually of the "fly-ball" type. The piston speed depends on the r. p. m. and the stroke.

MEDIUM SPEED ENGINES have rotative speeds and strokes intermediate between the foregoing. Positively driven multiple valves are generally used and the cutoff is often operated by a separate valve. The governor is nearly always of the "shaft type." The piston speed is around 600 feet per minute, being higher on the larger engines.

The medium and low speed engines are usually of larger power than the high speed.

There is no sharp dividing line between these different types of engines and it is sometimes difficult to decide in which class an engine belongs.

31. AREAS OF VALVE OPENINGS. Of course, from one standpoint, the larger the openings the better will be the results obtained, for the less will be the throttling, or "wire-drawing" of the steam. But with larger openings there will be greater weight and cost of parts, increased clearance volume, and greater frictional and inertia forces of the valve gear, which latter not only decrease the mechanical efficiency of the engine but also disturb the action of the shaft governor, if one is used. Therefore the problem before the designer is to determine how small it is advisable to make the openings.

For the rate at which the steam is supplied to the cylinder to be always equal to the rate at which the volume is displaced by the piston, the following expression must be satisfied:—

$$av = AV \qquad \ldots\ldots \qquad (1)$$

in which a = area of passage (sq. in., usually)

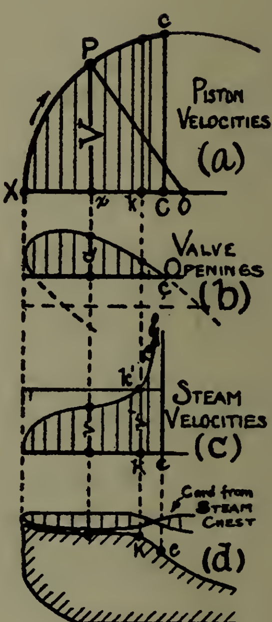
A = " " piston (same unit)

v = velocity of steam (ft. min., usually)

V = " " piston (same unit)

Then $$v = AV \div a \qquad \ldots\ldots \qquad (2)$$

$$a = AV \div v \qquad \ldots\ldots \qquad (3)$$



FIG·37

Applying equations 1, 2 and 3 to the flow of steam through the opening of the valve, A is the only constant quantity in them. How the other quantities, a, v and V, vary, will now be shown, the valve opening considered being that of the steam edge of the head end.

Assuming that the piston has simple harmonic motion, it will be remembered that for such motion, if the velocity of the crank pin is represented to some scale by OP, the radius of the crank pin circle, Fig. 37a, then the ordinate of the crank pin Px represents the velocity of the piston to the same scale. Under these conditions the **Curve of Piston Velocities** is a circle, the ordinates being the velocities and the abscissae the positions of the piston. If any other scale is used for the ordinates, the curve becomes an ellipse. If the cutoff takes place when the piston is at C, the ordinates during the period of admission are those under the arc XPc.

A **Diagram of Valve Openings** is shown in Fig. 37b for the steam edge of the valve, the ordinates being areas of openings, and the abscissae the piston positions as before. This diagram is really that part of the Elliptical Diagram (see Fig. 13) which lies above the steam lap line, the scale being changed to read areas instead of widths of openings.

Now since we know A, the area of the piston, and have curves giving simultaneous values of a and V, then from equation (2) the various values of v may be computed. If these are plotted as ordinates on piston positions as abscissae, the **Curve of Steam Velocities**, Fig. 37c, is obtained.

It is seen that the steam velocity is comparatively small in the early part of the stroke, but rises to infinity when the valve closes, the curve becoming assymptotic to the ordinate at C.

Now, when the opening becomes very restricted, and the velocity of fluid is very great, the throttling, or "wire-drawing," will prevent a nonexpansive fluid, such as water, from following up the piston and therefore the pressure in the cylinder will fall off. The velocity at which this failure of the working fluid takes place will be termed the **critical velocity** and is shown by the ordinate v′ in Fig. 37c, the piston being at K. In the case of an expansive fluid, however, such as steam, after this point is reached the fluid which is already in the cylinder, and the little which still manages to enter, expands to fill the space behind the piston so the pressure drops much more gradually than in the former case.

By comparing the indicator cards taken from the cylinder with those taken simultaneously from the steam chest, as in Fig. 37d, we may determine K, the position the piston occupies at the time when the admission line begins to round off towards cutoff. Then computing the velocity of the steam through the effective opening of the valve, corresponding to this piston position, the critical velocity is determined. From data obtained in ths way from several engines of different types the critical velocity of steam was found to be about 14,000 feet per minute, by Professor M. E. Gutermuth, of the Technical High School of Darmstadt.*

Each kind of valve has its typical curve of steam velocities, and those which cause K to be nearer to C (Fig. 37c) of course give the sharper cutoffs. In the case of the simple slide valve, the maximum opening of the valve will be considerably more, and the corresponding velocity of the steam, proportionately less than the critical.

32. VALVE OPENINGS USED IN DESIGNING. The **Gutermuth Method** of determining the proper valve openings is as follows:— First having selected the point on the indicator card where the rounding towards cutoff is to begin, note the position of the piston and determine its velocity. Then using equation 3, compute the area of the **critical opening**, for v equal to 14,000 ft. per minute or less. The valve gear can now be designed to give this opening when the piston is in the position previously selected, and to give openings corresponding to velocities less than the critical, before this point is reached.

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*See Journal Am. Soc. Naval Engineers, May 1904.

The **usual method** followed in the past has been to entirely disregard the critical opening, and design the valve to have a **maximum opening** which corresponds to a velocity of steam which has been found by experience to give satisfactory results. The maximum area of opening is found by using equation (3); but V is the **mean velocity of the piston** (2LN), and v is the **"mean velocity of the steam."** In practive v varies from 6,000 to 10.000 feet per minute, but it is usually in the neighborhood of 8,000

The shorter and more direct the passages, and the more rapid the opening and closing of the valve, the higher may be the velocity allowed. When it is desired to have an especially sharp cutoff, the valve is given wider opening (corresponding to a low velocity), the edge sometimes even overtraveling the back edge of the port. This is of course at the expense of having to use somewhat larger parts in the valve gear.

As the steam lap is always a great deal larger than the exhaust lap, on a D-valve, the opening of steam is much more restricted than that to exhaust. In designing the gear it is only necessary to see that the steam opening is sufficiently large, for the exhaust opening will always be more than is required.

The **width of the valve opening** is of course equal to the area divided by the length, which is nearly always equal to that of the port.

33. AREAS OF PORTS AND PASSAGES. The area of the piston and that of the passage are constant qualities, so, if the passage were always open, the velocity of the steam through it would be directly proportional to the velocity of the piston.

In determining the area of a passage, equation (3) is used, but it is customary to use for **V the mean velocity of piston** and for **v a "mean velocity of steam"** which has been found by experience to give satisfactory results.

In 1895, under the direction of Professor John H. Barr, data relating to the proportions of engine parts was obtained from a large number of manufacturers of standard commercial types of stationary engines.* This collection of material covered nearly two hundred simple engines of sizes ranging from 20 to 750 horsepower. From these data the mean velocities of steam through the passages were computed. In the following table are given the results, including the range of velocities found and the values which appeared to be the mean practice, the latter being in bold type:—

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*"Current Practice in Engine Proportions" by J. H. Barr, Trans. A. S. M. E., Vol. XVIII, 1897.

MEAN VELOCITY OF STEAM IN FEET-MINUTE. (BARR).

Type of Engine	High Speed	Corliss
Steam pipe	5800—**6500**—7000	5000—**6000**—8000
Exhaust pipe	2500—**4400**—5500	2800—**3800**—4700
Exhaust passage	4500—**5500**—6500	4000—**5500**—7000
Steam passage	(Same passage)	5000—**6800**—9000

It is seen that the values used in practice vary over a wide range.

It will be remembered that on "single valve" engines, which class includes nearly all the high speed engines, the same passage is used for both admitting steam to and exhausting it from the cylinder, and therefore must be made large enough for the latter function.

For mean piston speeds of about 600 ft. min., which is quite common practice, the proportions will be about as given in the following tables:—

Area of Passage ÷ Area of Piston		
Type of Engine	High Speed	Corliss
Valve opening to steam	$\frac{1}{13}$	$\frac{1}{11}$
Exhaust Passage (cyl.)	$\frac{1}{13}$	$\frac{1}{11}$
Steam Passage (cyl.)	$\frac{1}{10}$	$\frac{1}{9}$
Diameter of Pipe ÷ Diameter of Cylinder		
Type of Engine	High Speed	Corliss
Steam Pipe	$\frac{1}{3}$	$\frac{1}{3}$
Exhaust Pipe	$\frac{4}{10}$	$\frac{4}{10}$

From Seaton and Rounthwaite's "Pocket Book of Marine Engineering" the following is quoted as applying to marine engines:—(H. P., M. P. and L. P. refer respectively to the high, mean, and low pressure cylinders of triple expansion engines.)

"The following figures give such speeds of steam as are usual in triple engines of the best class; but it must be understood that in dealing with very high piston speeds (say over 900 feet per minute), it is not always either possible or advisable to give such large areas:

SPEEDS OF STEAM.

Main steam pipe, 8,000 feet per minute; or say 8,100 feet, then—

$$\text{Diameter} = \frac{\text{Dia. of H. P. cyl.}}{90} + \sqrt{\text{Mean piston speed}}$$

Mean of maximum valve openings.	H. P.	7,500 ft. per min.		
	M. P.	9,000	"	
	L. P.	12,000	"	
Ports (during exhaust).	H. P.	5,800	"	Nearly equivalent to 40, 50, and 60 c. ft. of cylinder per sq. inch of port per minute.
	M. P.	7,200	"	
	L. P.	8,600	"	
Exhaust pipe or passage from one cylinder to next or to condenser.	H. P.	4,500	"	
	M. P.	5,500	"	
	L. P.	7,000	"	
Ports (during exhaust) in light, high-speed engines.	H. P.	5,800	"	Nearly equivalent to 40, 60, and 80 c. ft. of cylinder per sq. inch of port per minute.
	M. P.	8,600	"	
	L. P.	11,500	"	

For **Two-stage** or **Compound** engines use the above figures, only omitting those referring to M. P. cylinders."

In the case of multiporting, the aggregate area is usually made a little larger (say 10 per cent.) than for a single port.

The **lengths and widths of the ports** are usually the same as those of the passages, but when the valve has an auxiliary port, as in the Sweet and some other valves, the widths must be increased, as will be seen later.

The **effective width of opening** is of course equal to the area divided by the length.

The **length of opening** for flat slide valves is from 8-10 to once the diameter of the cylinder, the former value being used when the opening of the steam pipe into the steam chest is so located that a wider valve would obstruct the passage. On Corliss engines the length is usually the same as the diameter of the cylinder. In the case of piston valves the length of port is the

circumference and this is made from 25 to 60 per cent. greater than the diameter of the cylinder, the latter value corresponding to a diameter of valve (about) equal to half that of the cylinder. This increased length permits narrower width of opening and shorter valve travel; or when the opening is not very effective on that side of the valve which is away from the cylinder, as is sometimes the case, it provides an increased area. If the valve bushing has bridges across the ports, allowance must be made for the consequent reduction of tht effective length of port, which may be from 25 to 35 per cent.

34. CLEARANCES. The linear distance between the piston and cylinder head, when the former is at the end of the stroke, is called the **mechanical clearance.** One-eighth inch or more may be allowed for inaccuracy of parts and faulty initial adjustment, and from 1-32 inch to 1-8 inch may be allowed for the wear of each bearing (usually three) affecting the clearance. In the better class of stationary commercial engines the clearance is from ¼″ to ⅜″, the former value being used when both the piston and cylinder head have machined faces and the latter when this is not the case. On vertical marine engines from 50 to 100 per cent more clearance is allowed at the bottom of the cylinder than at the top, as the adjustment for wear in bearings decreases the space at the bottom.

The **clearance volume** is that space which must be filled with steam when the piston is at the end of the stroke, and includes the volume between the piston and the cylinder head and that in the passages out to the valve faces. It is usually expressed as percent of the volume displaced by the piston in one stroke. On stationary high speed engines it is usually from 6 to 13 per cent., on low speed engines from 2 to 6 percent. and intermediate between these values on medium speed engines.

On marine engines with flat valves the clearance for large cylinders is from 8 to 14 per cent. at the H. E. With piston valves the clearance is usually from 1½ to 2 times that of flat valves. Vertical marine engines have larger clearance at the bottom than at the top on account of the greater mechanical clearance.

Clearance is objectionable, but unavoidable, and should be made as small as possible. It may be reduced by making the passages shorter and more direct, and by reducing the mechanical clearance. For a given clearance volume and diameter of cylinder, the percentage of volume is less on long stroke engines than on short. With certain types of valve gear, as will be seen later, to prevent excessive compression large clearance is necessary.

35. DRAWING HYPERBOLIC EXPANSION LINES. First Method. Start with the X and Y axes, Fig. 38, and using absolute pressures and volumes as co-ordinates, locate the point of cutoff, c, and draw the vertical line c c′. Through any point 1 in the line cp produced, draw 01 from the origin, to intersect cc′ at

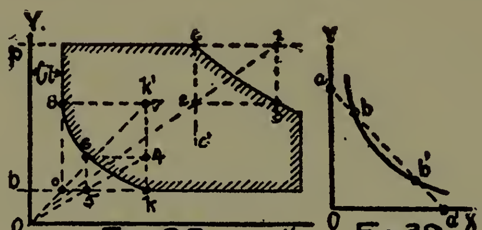


FIG. 38 FIG. 39

2. Then the vertical and horizontal lines drawn respectively through 1 and 2, intersect at 3, which is a point on the expansion curve. Other points may be found in the same manner.

Starting at k, the beginning of compression, draw the vertical line k k′. Through any point, 4, on this line, draw 04 from the origin to intersect the back pressure line kb at 5. Then the horizontal and vertical lines drawn respectively through 4 and 5, intersect at 6, which is a point on the compression line. The pressure at the end of compression is found by first determining the point of intersection 7 between Oo and k k′, and then from this point drawing horizontally to intersect the clearance line at 8.

Second Method. To construct a hyperbola through the point b, Fig. 39, draw any line through this point to intersect the axes at a and a′. Along this line from a lay off a distance equal to ab, locating the point b′, which is a point on the hyperbola.

36. CUSHIONING THE RECIPROCATING PARTS. First suppose there is no compression. Then when the piston approaches the end of its stroke the effective steam pressure and the inertia of the reciprocating part are both acting towards that end of the stroke, taking up the slack in the bearings of the reciprocating parts. Now, when the steam is admitted on the other side of the piston the pressure on the bearings is rtversed more or less suddenly. With reciprocating parts of small weight, and with high steam pressure, this reversal will be very sudden and if there is much "play" in the bearings, and there must always be a little, the consequent impact will not only produce noise but will cause excessive stresses in the impinging parts.

One method of preventing the occurrence of these undesirable features is to make the weight of the reciprocating parts

so great that their inertia will oppose the steam pressure sufficiently to cause the play in the bearings to be taken up gradually, thus preventing impact. But as the inertia forces are free forces which tend to move the engine on its foundation, it is usually desirable to have them small, even when counterbalancing is attempted, so this method is usually unsatisfactory.

Another method is to arrange the valve to open gradually, but this is accompanied by a more gradual cutoff, which is undesirable.

The best method is to gradually reverse the pressure on the bearings by introducing compression, then, when admission takes place there is no play to be taken up and consequently no impact in the bearings.

The **force due to the inertia of the reciprocating parts** is greatest when the crank is on the H. E. dead center, and, as the steam pressures used are expressed as intensities per square inch of piston, the inertia force will be expressed in the same way. Without going into the derivation of the formula, the maximum inertia force per square inch of piston is

$$h = \frac{wRN^2}{35,200}\left(1 + \frac{1}{n}\right) \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

in which w = total weight of reciprocating parts (in lbs.) divided by the area of the piston. This is from 2 to 3½ lbs., the larger values being for the longer strokes and larger engines.

R = radius of the crank (inches).

N = revolutions per minute.

n = ratio of length of connecting rod to crank radius

Now if V is the piston speed in ft. minute, then

$$12\ V = 4\ RN, \qquad \text{or} \qquad RN = 3\ V$$

Substituting this value of RN in equation (4) we get,

$$h = \frac{w\ V\ N}{11,733}\left(1 + \frac{1}{n}\right) \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

Then if the piston speed is about 600 ft. minute,

$$h = w\ N \div 16.8, \qquad \text{when } n = 6 \quad . \quad . \quad . \qquad (6)$$

$$= w\ N \div 15, \qquad \text{when } n = 5\tfrac{1}{2} \quad . \quad . \quad . \qquad (7)$$

37. COMPRESSION. The final compression pressure should not be greater than the initial steam pressure, and to cushion the reciprocating parts sufficiently to reverse the pressure on the bearings, it must overcome the pressure of steam on the

other side of the piston as well as the inertia force of the reciprocating parts. Expressed algebraically

$$k \overset{>}{=} b + h < p \quad . \quad . \quad . \quad . \quad . \quad (8)$$

In which
k = steam pressure at the end of compression.
b = " " on opposite side of the piston.
h = intensity of inertia force (equations 4 to 7.)
p = initial steam pressure.

Having found k it is still necessary to determine at what point in the stroke compression should begin to reach this pressure.

Let x = % of stroke for compression to begin
b = back pressure
c = clearance volume %

Then assuming a hyperbolic compression line and using absolute pressures,

$$k \times c = b \times (c + x) \quad . \quad . \quad . \quad . \quad (9)$$

or
$$x = \frac{k \times c}{b} - c \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The same thing may be found by graphical construction, of course.

When there is low back pressure, (such as occurs on condensing engines) accompanied by large clearance, it is sometimes not possible to get sufficiently high pressures at the end of compression, and then other expedients must be adopted for cushioning, such as employing a gradual lead, by having the edge of the valve oblique with that of the port, or by drilling through the valve small holes which open a little before the edge of the valve does.

Compression should never be sufficient to lift the valve off its seat. In the case of the variable eccentric, the Stephenson Link and other similar valve gears, the compression increases as the cutoff is reduced, so it is necessary to investigate the compression pressure corresponding to the earliest cutoff in this respect.

On vertical engine it is advisable to have the compression at the bottom of the cylinder greater than at the top, as, besides the inertia of the repicrocating parts, their weight must be counteracted.

38. RELEASE should occur from 2 to 15 per cent. from the end of the stroke depending on the speed of the engine.

As release and compression are conjugate events when operated by the same valve, if one is fixed, that determines the other. Often it is not possible to have both these events occur as desired, in which case a compromise must be made.

39. LEAD. As the clearance must be filled and as a large percentage of the steam is condensed by the walls of the clearance space when admission takes place, it is usually necessary to give the valve lead. The smaller the compression is and the greater are the clearance volumes and the speed, the larger must be the lead. When the compression is up to the initial steam pressure, very little lead is needed. Sometimes no lead, or even negative lead is used.

Another effect of lead is to increase the earlier valve openings, in the case of the variable eccentric and the link gears. This feature may be especially valuable, for when these gears operate at the earlier cutoffs the openings are very restricted.

It is difficult to give rules for lead as practice varies widely. For high speed engines it may be made 1-16 inch for each inch of radius of the eccentric, which corresponds to a lead angle of about 3½ deg. (slope 1:16), although double this amount is sometimes used. On large marine engines with large clearances the angle is even as large as 12 deg. at the H. E. On vertical marine engines the lead at the bottom is usually from 50 to 100 per cent. larger than at the top, to make provision for the decrease in lead at this end due to wear of valve gear parts, and to fill the larger clearance volume at this end of the cylinder.

40. CUTOFF. When a simple D-valve is used, cutoff should occur not much before ⅝ stroke. (See **28.**) On high speed engines, on which the early compression and release are not so objectionable, and on which special valves are used, the normal cutoff is usually at ¼ stroke, with a range from 0 to ½ or ¾. The cutoffs for medium speed engines are about the same as for high speed. On Corliss engines the normal cutoff is from 1-5 to 1-4 stroke, with the tendency toward the former value for single eccentric gears and toward the latter when double eccentrics are used. In the case of link gears the latest cutoff is from ⅝ to ⅞ stroke, usually, and the normal cutoff is later than ½ stroke.

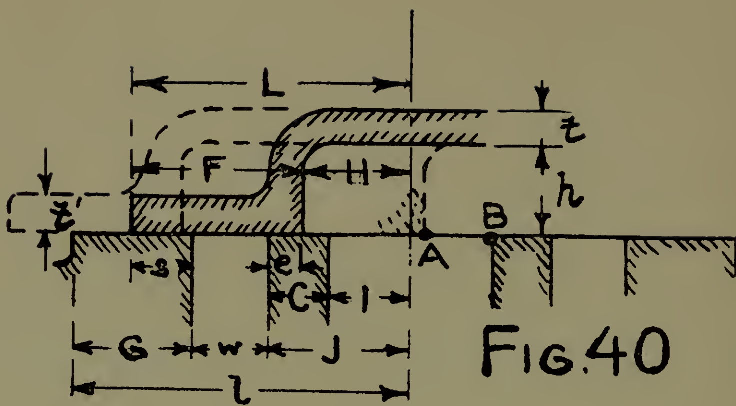


FIG. 40

41. DRAWING A D-VALVE. It is better to draw the valve in its central position, as shown in Fig. 40, for the H. E. The edges of the valve, of course, overlap the edges of the ports by amounts equal to the laps. To provide proper passage for the exhaust steam the height h of the cavity in the valve must not be less than the width of passage w; and when the valve is to the extreme right, the distance AB should be at least equal to w. The bridge C should be wide enough for steam tightness when the valve is to the right and to have sufficient surface to prevent wear. It is not usually less than the thickness of the cylinder. G should be wide enough to prevent the exhaust edge from overtraveling when the valve is to the extreme left and should be narrow enough to let the steam edge of the valve overtravel some, so as to prevent wearing a shoulder.

Rough Rules for the thickness of metal in the valve are

$$t = \tfrac{1}{2}T \text{ to } \tfrac{5}{8}T \qquad (11)$$

$$t' = .8T \text{ to } .9T \qquad (12)$$

in which T is the thickness of the cylinder wall, and t and t′ are the thicknesses shown in Fig. 40. For stationary commercial engines Professor Barr gives

$$T = .05\,(\text{cylinder diameter}) + .3 \qquad (13)$$

the unit being inches.

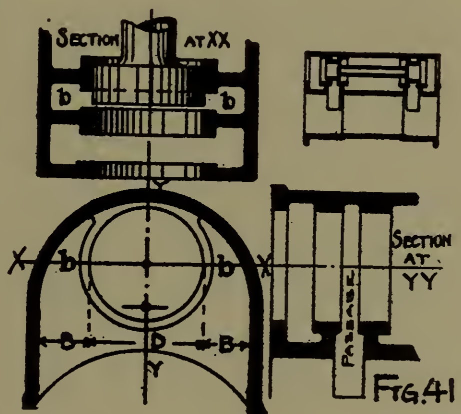
If the valve is not "balanced," it is desirable to make it as small as possible, so as to reduce the friction and weight to the minimum. Consequently the valve is laid out to give the narrowest width of exhaust cavity and this may be done in the following manner:—

Having drawn the valve seat and having determined the face of the valve F at one end of the cylinder, say the H. E. as in Fig. 40, move the valve to the extreme right and lay off the

proper distance from its exhaust edge A to locate B. The drawing of the valve and seat can then be completed readily.

When the valve is central its center line coincides with that of the valve seat, which is usually made symmetrical. If the laps of the two ends are not equal, the valve will be unsymmetrical. Then in determining the shortest allowable exhaust cavity that end of the valve which has the greatest exhaust lap (algebraically) is the one to be considered.

Piston Valves may be considered to be flat valves bent into cylindrical form. They are laid out in just the same manner as flat valves.



Looking at the end view in Fig. 41, it is seen that the steam, which issues from the upper semi-circumference of the valve, must go through the passages bb to reach the cylinder. The effective areas of these passages b are shown in the section in the upper left hand corner of the Figure. Their combined area should be one-half that needed for the exhaust.

If a bush is used for the valve seat, its thickness may be

$$t = 3/8'' + D/30 \qquad . \quad . \quad . \quad . \quad . \qquad (14)$$

in which D is the diameter of the valve in inches. If bridges are used across the port, their width may be made a little greater than their thickness (say 20 per cent.) and, to distribute the wear more evenly, they may be made oblique (usually 60. deg.) with the edge of the valve.

Internal Valves, whether flat or piston, may be "laid out" in a similar manner, the only difference being that the steam and exhaust laps change places.

PART II

VALVE GEAR FOR HIGH SPEED ENGINES

MULTIPORTED AND BALANCED VALVES
SHAFT GOVERNORS
VARYING ECCENTRIC

CHAPTER I.

INTRODUCTION.

42. VALVE GEARS FOR HIGH SPEED AUTOMATIC ENGINES.—(a) The High Speed type of engine was briefly described on page 40. In general, these engines have automatic regulating devices which cause them to develop just enough power to meet the demand and at the same time maintain approximately constant rotative speeds at all loads. The regulation is accomplished by a "Shaft Governor," which controls the point of cut-off by varying the position of the eccentric with respect to the crank.

b. The **"range of cut-off"** is usually from 0 to ⅝ or ¾ stroke, and the **"range of load"** is from "friction load" to 50 per cent (and sometimes even 100 per cent) "overload," this latter being with respect to the "normal load" or load at which the engine usually operates. The normal load should of course be that at which the engine gives the best economy in the consumption of steam, which, for simple engines, corresponds to a cut-off at about ¼ stroke.

On page 39 it was seen that if a simple slide valve is used to give a cut-off as early as ¼ stroke, there will be introduced certain features which are usually undesirable. How these features can be overcome or be used to advantage in this type of engine, will now be considered.

c. It was seen that the early cut-off is accompanied by an **early compression** which results in a terminal pressure which is ordinarily objectionably high. This terminal pressure can be reduced, however, by increasing the ratio between the clearance volume and that displaced by the piston. With a given clearance volume and piston area, this clearance ratio may be made large by using a short stroke. Hence, engines of the type under consideration are made with relatively **short strokes** (usually from one to one and a third times the diameter of the cylinder).

Q201. Given, clearance volume 300 cu. in., diam. of cylinder 16 in., stroke 36 in., back pressure 17 lbs. absolute, and compression beginning at 70% of stroke.

(a) What is the per cent. clearance volume?

(b) What is the pressure at the end of compression?

With the same data except that the stroke is 18 in.,

(c) What is the per cent. clearance volume?

(d) What is the pressure at the end of compression?

If an engine runs at a high rotative speed, a fairly high compression, instead of being a fault, may be a necessity, in order to properly "cushion" the reciprocating parts. Further, with this high speed, the early release (which accompanies the early cut-off) is a desirable feature, whereas on a low speed engine it would be objectionable. Hence these engines are not only "short stroke," but are also **"high speed."**

Q202. Given, the weight of the reciprocating parts, per sq. in. of piston, as 2½ lbs., stroke 36 in., R.P.M. 100, and length of connecting rod equal 6 cranks.

(a) What is the inertia force of the reciprocating parts (per sq. in. of piston) at the head end of the stroke? (See §36).

(b) Same except with stroke 18″ and R.P.M. 200?

(c) Is the terminal pressure found in Q201d sufficient to overcome this inertia?

d. The **excessive size of the valve gear parts,** which ordinarily occurs when a valve is designed for an early cut-off, may be overcome by increasing the length of port which calls for a narrower valve opening and a corresponding reduction of the laps, travel, and size of the eccentric; and these in turn are accompanied by a decrease in the friction and wear of the valve gear. The greater length of port may be obtained by using a wide valve, or by using a **multiported valve.**

With the type of valve gear which is used on this class of engine, the travel of the valve varies with the cut-offs and the earlier the latter are, the more restricted are the openings of the valve. When the openings are ample for the latest cut-offs, **auxiliary ports** may be added to the valves in such a manner as to assist during the early openings only, and to have little or no effect on the wider openings.

Sometimes **special kinematic arrangements** of the valve gear are employed to give wide openings with small travel, but this is at the expense of complication of mechanism. Examples of these various types of valves will be given later.

e. The **friction of the valve** is undesirable not only because it decreases the mechanical efficiency of the engine, and causes wear, but also because it disturbs the action of the shaft governor. The work done against friction may be reduced by decreasing the motion and by lessening the pressure between the face of the valve and its seat. The pressure of the valve on its seat may be due in part to the weight of the valve, but it is principally caused by the steam pressure. The back of the valve is subjected to the pressure of the live steam, whereas a part of the face is exposed to the exhaust pressure, the resulting or unbalanced pressure forces the valve against the seat causing the friction and wear. To reduce this unbalanced pressure to a minimum, various schemes of "**balancing**" the valve are employed. The simplest scheme is to use a piston valve, which is subjected to equal pressures on opposite sides and is therefore balanced because of its shape. Other means of balancing, which will be described later, are to use "balance or equilibrium rings," "balance or pressure plates," and by using a "floating valve."

f. On vertical engines when the **weight of the valve gear** is great, there are sometimes attached to the upper end of the valve stem "balance pistons," which are of such size that the steam pressure on the bottom side will carry the weight of the valve gear parts.

g. The **inertia** of the valve gear parts, which adds to the friction and wear and disturbs the action of the governor, may be reduced by making the valve and gear of light weight.

h. In case there should be water in the cylinder during the compression phase, in quantities more than sufficient to fill the clearance volume, **relief** of some kind must be offered, otherwise some part of the engine must break or yield. "Relief Valves," which are somewhat similar to boiler safety-valves, are frequently used, but often the slide valve itself may be arranged to lift off of its seat and afford sufficient relief.

SUMMARY. It is evident from the foregoing discussion that one must be familiar with the arrangements and methods of operation of shaft governors, with the special types of valves, and with the action of the variable eccentric before one can undertake the design of a valve gear for a high speed engine. These features will therefore be considered next, but only the more typical cases will be taken up. The student should supplement the text by a study of the catalogues and drawings of the various manufacturers.

CHAPTER II.

TYPES OF VALVES USED ON HIGH SPEED ENGINES.

43. In studying the valves discussed in this chapter note the following features:

(1) Relative Sizes and Weights (inertia effect).
(2) Methods of Multiporting and Auxiliary Porting.
(3) Methods of Balancing.
(4) Provision for Water Relief.
(5) How wear is taken up and provision for re-machining.

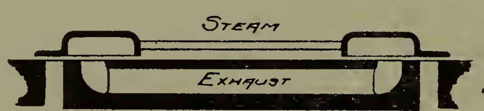


Fig 42

44. **SIMPLE DIVIDED VALVE.** If a cylinder is long and has direct passages, the valve as ordinarily constructed would be of excessive size and weight. Fig. 42 shows how these undesirable features may be avoided by making the valve in two parts.

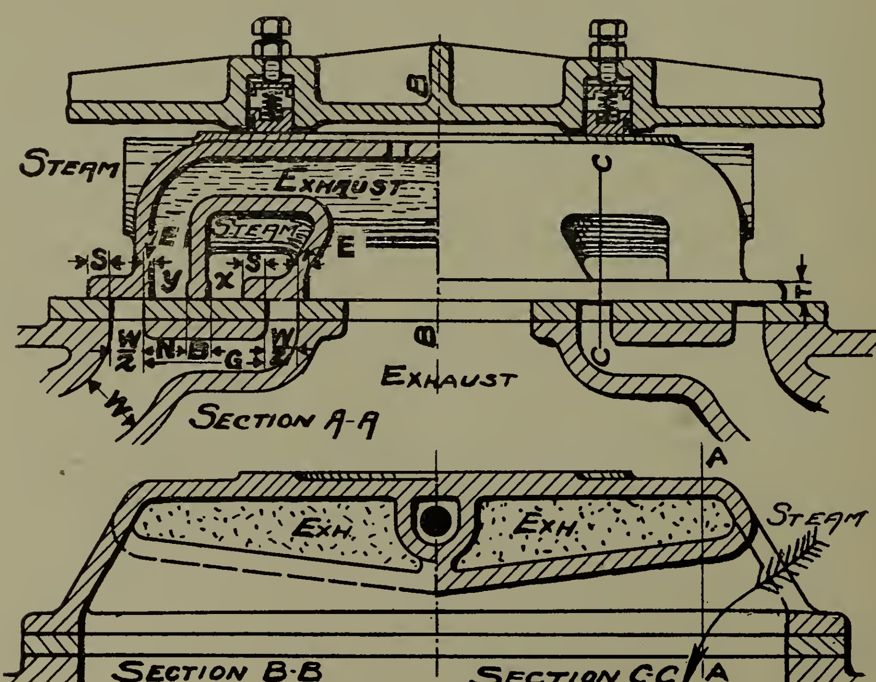


Fig 43

45. DOUBLE PORTED MARINE VALVE. Figure 43 shows a valve which is double ported to both the steam and the exhaust. The steam enters the cylinder past the end of the valve and also through the port x, which latter is supplied through a passage opening from the side of the valve. The exhaust takes place past the inner edge and also through the port y.

(a) If the back of the valve is subjected to the pressure of the steam, and a part of the face of the valve is exposed to the exhaust pressure, the resulting unbalanced force presses the valve against its seat causing friction and wear as movement takes place. In order to reduce these undesirable features, it is necessary to protect a sufficient area of the back of the valve from the steam, to "balance" the valve. In this case this balancing is accomplished by employing a steam tight "**balance or equilibrium ring,**" which is fitted in the steam chest cover and held against the back of the valve by springs. The space inside this ring is open either to the atmosphere or to the exhaust. The valve should not be completely balanced, for it should press against the valve seat with sufficient force to maintain a steam tight joint. In case there is water in the cylinder this valve can lift off of its seat and afford a relief. The valve is here shown to ride on a "**false valve seat,**" which can be replaced when worn. Also the metal at the face of the valve is made with extra thickness to allow for remachining.

(b) **Proportions.** Since the valve is double ported, the portwidths, laps and travel will be only half those which would be used on a single ported valve. In Fig. 43 the steam and exhaust laps are S and E. The passage W must of course be wide enough for the exhaust steam. B should be wide enough to maintain a steam tight joint, and G should be equal to the travel of the valve plus B, in order that B will not overtravel into either port. When the valve is central, B should be in the middle of G. The thickness of metal of the valve may be made the same as for a simple valve. (See page 52). The thickness of the False Valve Seat may be made the same as that of the cylinder wall or greater.

X201. Draw a section of a Double-Ported Marine Valve, in its central position, making S 1 in., E ⅛ in.. W 1½ in., B ¾ in., throw 1¾ in. Only one end of the valve need be drawn.

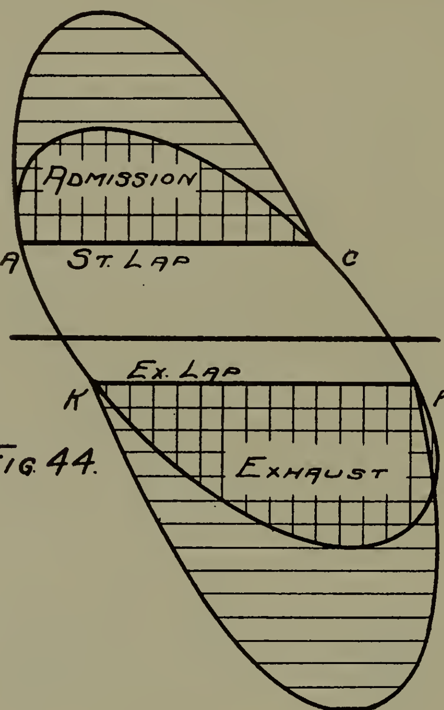


FIG. 44.

(c) In the **Elliptical Diagram** (Fig 44) the ordinates of the portion which is sectioned vertically represent the openings of a single ported valve. The openings occuring when the valve is double ported are shown by the ordinates of the areas which are sectioned horizontally.

The double-ported openings can be shown on **the Zeuner and the Sweet Diagrams** in a somewhat similar manner as shown by curves A. 9 C. in Figs. 57 and 58.

X202. Given diameter of cylinder 18 in., stroke 24 in., R.P.M. 150, length of port 18 in., cut-off ¾ stroke, lead ⅛ in., and compression beginning at 85% of stroke. For a Double-Ported Marine Valve—(a) Compute the width of opening to steam, using 8,000ft. min. as the velocity of the steam. Also determine the width of port using 5,500 ft. velocity of the exhaust.

(b) Draw a Bilgram Diagram.

(c) Draw a longitudinal section of the valve.

(d) Draw an Elliptical Diagram, showing the double-ported action.

(e) Draw Zeuner and Sweet Diagrams, and on them show the double openings.

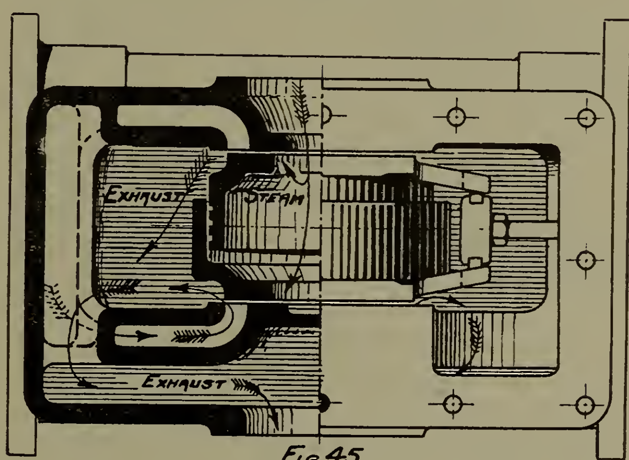


Fig. 45

46 BALL TELESCOPIC VALVE. The valve, shown in Fig. 45, is really composed of two valves, each having its own seat, and each having on its back a cylindrical portion which telescopes with the similar part on the other valve, the joint of course being steam tight, but permitting relative movement. Steam is admitted into the middle of the valve and is exhausted at the ends. The valve is practically balanced, as each part acts as a balance ring for the other; however, there is enough unbalanced pressure to maintain steam tight joints at the valve seats and also to take up the wear. Exhausting at the ends has the advantage that the stuffing-box for the valve stem, is subjected to the pressure of the exhaust steam only and is therefore much easier to keep tight. In case there is water in the cylinder the parts can lift off of their seats to give relief. The ports are necessarily long and circuitous and the clearance volume large.

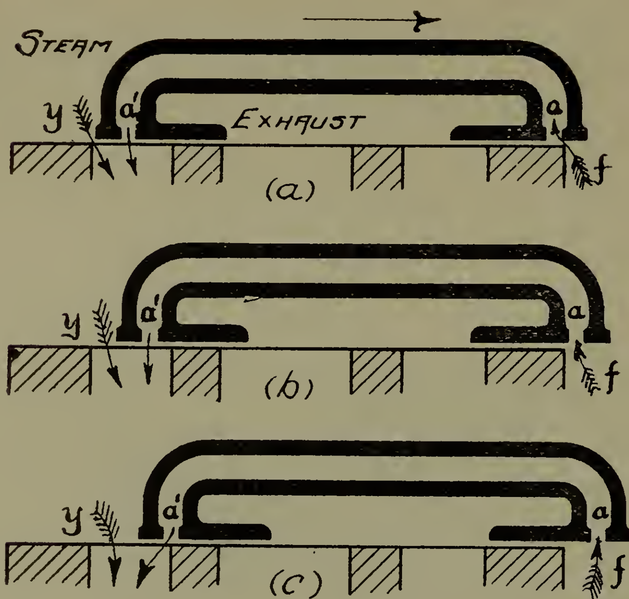


FIG 46

47. ALLEN OR TRICK VALVE. This valve has an auxiliary passage aa′ (Fig. 46a), so arranged that, as the valve moves to the right, it opens at f at the same instant that the main steam edge opens at y to admit steam. The exhaust is single ported.*

(a) Considering the valve as moving to the right, the phases of opening of the steam edge are as follows: (1) The

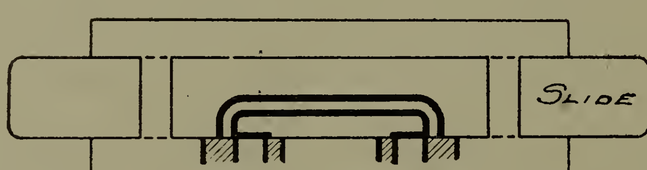


FIG. 47

* The student is advised to construct a model of a symmetrical valve similar to that shown in Fig. 47, using two pieces of stiff paper, one piece 3 inches by 6 inches and the other 1 inch by 8 inches. Referring to Fig. 50, for dimensions use W ⅜ inche, N 1-16 inch, E ⅛ inch, S and S′ 5-16 inch, B ⅛ inch, C ¼ inch and I 1¼ inches. This model will assist greatly in gaining an understanding of the further discussion.

double ported action (Fig. 46a), caused by y and f opening at the same rate, continues until a is wide open. **(2)** With the movement continuing, (Fig. 46 b), the opening at y continues to increase while that through a remains constant, i.e., the opening is "**single ported plus a constant.**" This phase continues until a′, the left end of the auxiliary passage, begins to be throttled by the exhaust edge of the valve seat, as in Fig. 46c. **(3)** With further movement to the right, a′ is throttled at the same rate as the main steam edge opens, the effective opening remaining **constant** until the auxiliary passage is entirely closed. **(4)** After this the opening is **single ported** until the cylinder port is wide open.

Now if the valve moves to the left to close, the effective opening will decrease with these phases occuring in the reverse order.

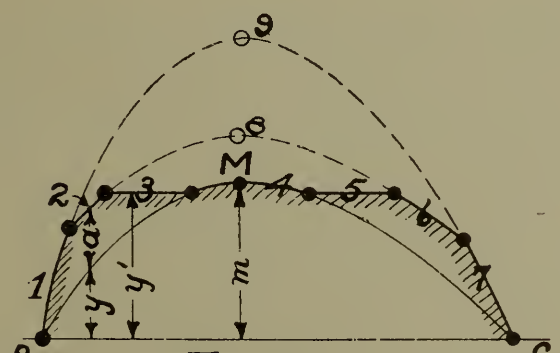


Fig 48

(b). A **Diagram of Openings** for the steam edges is given in Fig. 48, the elliptic curve AMC* showing the opening of edge y alone, and 1—2—3—4 etc. showing the total effective opening. If y is the ordinate of the former curve and y′ that of the latter, then, taking the phases in order, as numbered in the preceding paragraph, the values of y′ will be as follows: **(1)** $y' = 2y$; **(2)** $y' = y + a$, where a is the constant opening through the auxiliary port; **(3)** $y' =$ constant. $= W - B$, where W is the width of port and B is the width of metal at the end of the valve, (see Fig. 50); and **(4)** $y' = y$. The maximum opening of the main steam edge is m.

The **Zeuner** and **Sweet Diagrams** are indentical with those shown in Figs. 57 and 58 and are described in Par. 51h.

(c) **Case I.** If $W \lessdot B + m$, the case is that which has already been discussed.

* Purposely shown distorted.

(d) **Case II.** If W = B + m, lines 3 and 5 will be tangent to AMC at M.

(e) **Case III.** If W = B + m + a, curves 2 and 6 will meet at 8.

(f) **Case IV.** If W = B + (the total desired opening w′), and if a = m = ½ w′, then, the opening is **double ported throughout,** and curves 1 and 7 will meet at 9 with ordinate 2 m. It would appear at first that, since the steam opening is double ported, the steam lap and travel could be made half those required for a simple valve; but, as the exhaust is single ported, the opening of this edge may then be too restricted. Hence, it is necessary to find the throw which will give the desired opening to steam, and also that for the proper exhaust opening, and then use the larger of the two.

X203. Given:—Exhaust opening 1¾ inches; total steam opening at least 1¼ inches; lead ⅛ inch; C. O. ¾ stroke; compression 85% stroke; Allen valve with width of auxiliary port half the required opening to steam.

Required:—Throw, angle of advance, steam and exhaust laps for the head end of the valve. Draw the Diagram of Openings.

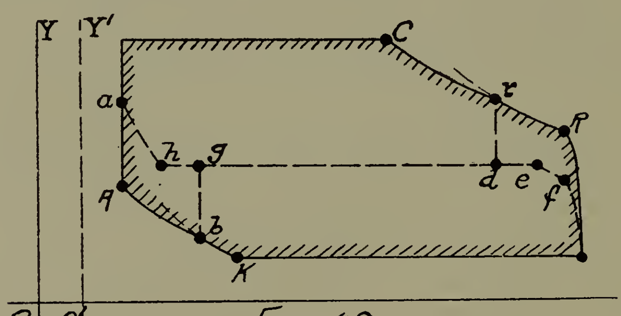


Fig. 49

(g) Considering the head end of the cylinder, after the valve has moved to the left far enough to cut-off the steam at y, (Fig. 46), the auxiliary port, now closed at f, is still open to the head end of the cylinder at a′, hence the expansion takes place with a **clearance volume** equal that of the cylinder plus the volume of the auxiliary passage. After this passage is closed by the valve moving farther to the left, the expansion is with the clearance volume of the cylinder alone. Fig. 49 shows an **indicator card** on which the closing of the auxiliary port is shown at r. After exhaust closure, the compression is first into the clearance of the

cylinder, and later it is also into the passage in the valve. In Fig. 49, the opening of the auxiliary port occurs at b.

OY is the axis for expansion and compression when the auxiliary passage communicates with the cylinder; and O'Y' is the axis when expansion takes place in the cylinder alone. The horizontal distance between the axes represents the volume of the auxiliary passage. Hyperbolas Cr and Ab are with respect to axis OY, while rR and bK are drawn with O'Y' as axis.

(h) In the foregoing paragraph it was assumed that the exhaust edge closed before the auxiliary port opened, i.e., that G is greater than W in Fig. 50. If G is just equal to W, then on the indicator card r will coincide with R and b with K. Unless G is equal or greater than W, release will occur prematurely through the auxiliary passage into the other end of the cylinder, which is still open to exhaust at this time.

(i) **Negative auxiliary lap** (N and n, shown positive in Fig. 50) may be used if the valve has positive exhaust lap, but from the preceeding paragraph (—N) must not be greater than (+E) nor (—n) greater than e. From the time the auxiliary port opens at one end until it closes at the other, the valve movement is N plus n, and during this time there is communication between the two ends of the cylinder with a resulting equalization of pressure. Fig. 49 shows by the broken lines the changes in the indicator card for this case. The communication is established when expansion has reached r, and compression b, the line of equalized pressure being d-g. Communication ceases at e and h. It is seen that the cushioning effect is increased with this arrangement and sometimes this is a very desirable feature, as in the case of a condensing engine in which conditions are such as to limit the compression to less than the desired amount.

> **X204.** Using the data of X203, and with negative auxiliary lap ⅜ inch, find the piston positions for the beginning and closing of communication between the two ends of the cylinder and draw a theoretical indicator card, with clearance of cylinder 4% and in auxiliary passage 4%, steam pressure 100 lb. gage, back pressure 2 lbs. absolute. Neglect the angularity of the connecting rod.

(j) **Proportions:** For notation see Fig. 50.

w'—is the desired width of opening to steam (total).
w—is the desired width of opening to exhaust (total).
m—is the maximum width of opening of the main steam edge.
a—is the width of the auxiliary passage.

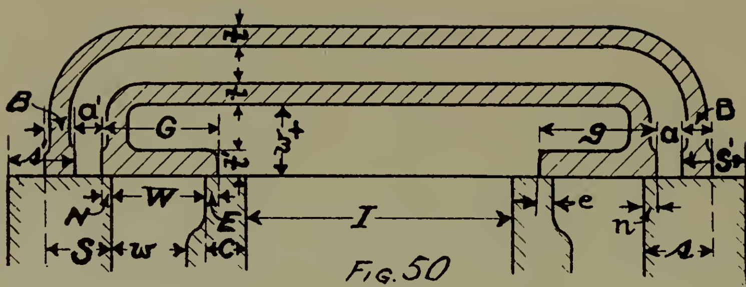


Fig. 50

If the auxiliary port does not increase the maximum opening of the valve, but merely augments the initial and final openings (as in Cases I. and II.) the throw, lead and laps will be the same as those found for a single ported valve. The amount of opening during the third phase (that of constant opening) may be arbitrarily assumed at any suitable value k; then W=k+B, but must, of course be greater than w.

The width of the auxiliary port (a) may be made anything within reasonable limits. It must be wide enough to permit the use of a core of thickness sufficient to withstand the flow of the molten metal when the casting is poured, and for this, the width should usually be somewhat greater than the thickness of metal in the valve.

To receive the full benefit of the auxiliary port during the whole of the period of opening, W must be at least equal to B+m+a, as in Case III.

To have double ported action during the whole period of opening, not only must W be equal to B+w′, but (a) must equal ½ w′. The throw would be that of a double-ported valve, provided it gives sufficient opening to exhaust, as in Case IV.

In addition, it must be seen that:—

$$S' = S \text{ and } s' = s$$

$$G \text{ and } g \geqq W$$

The thicknesses of metal may be the same as for a simple valve. (pg 52) or thinner if well ribbed.

X205. Given, diameter of cylinder 18 in., in., stroke 24 in., R.P.M. 150, length of port 18 in., cut-off ¾ stroke, lead ⅛ in., and compression 85% of stroke. For an Allen valves, which is to have double-ported openings to steam during the whole period of opening:

(a) Compute the widths of openings to steam

and exhaust, using for velocities of steam 8,000 and 5,500 ft. min., respectively.

(b) Draw a Bilgram Diagram.

(c) Draw the longitudinal section of the valve.

(d) Draw a Diagram of Openings for the steam edges.

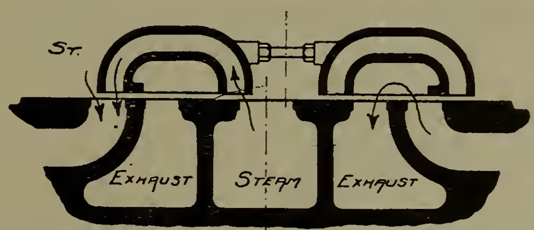


FIG. 51

48. DIVIDED ALLEN VALVE. This is shown in Fig. 51 and is the equivalent of the ordinary Allen Valve.

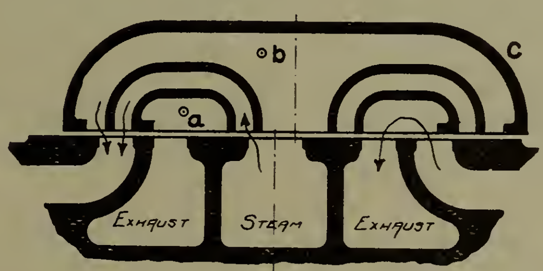


FIG 52

49. GIDDINGS FLOATING VALVE. This valve is obtained by adding the hood C in Fig. 52 to the divided Allen Valve of Fig. 51. The hood in no way affects the valve events; it is added for the sole purpose of balancing the valve. When the steam is first admitted inside the hood, the valve lifts from its seat letting steam into the steam chest until the pressure there is sufficient to reseat the valve. The valve then tends to float but the pressure on the back is always sufficient to keep it seated. To avoid the lifting of the valve, which lets some of the steam escape to the exhaust, little "needle valves" (a) and (b) are added in such a way as to maintain the proper pressure on the back of the valve to balance it. One of these needle valves connects the outside of the valve with the exhaust and the other with the steam passage.

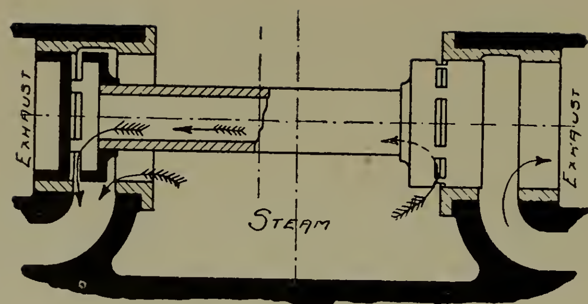


Fig 53

50 **ARMINGTON and SIMS VALVE.** Piston valves can be made like the Allen Valve. Fig. 53 shows the Armington and Sims valve which is of this same type, but is arranged to take steam at the middle and exhaust at the ends.

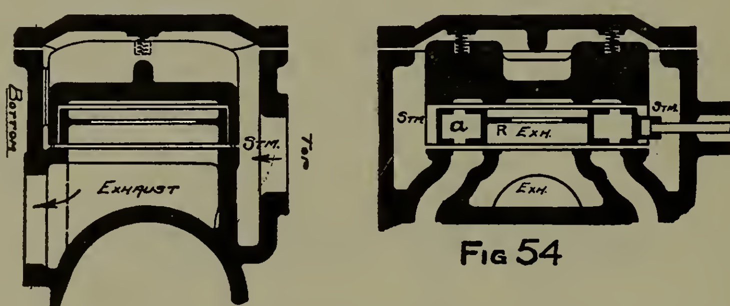


Fig 54

51. **SWEET VALVE.** Fig. 54 shows an "auxiliary-ported" valve which is of the "pressure-plate" type used by Prof. John E. Sweet on his Straight Line Engine. The valve, which is sometimes called the "Straight Line Valve," is really a rectangular piston valve which slides between the valve seat and a balance-

plate, or pressure-plate, which latter is supported by top and bottom distance-pieces so as to just clear the valve. The valve is double-ported to steam and exhaust, at least, during part of the period of opening. All the sliding surfaces are accurately scraped so as to give sufficient clearance to permit the valve to move freely, but not enough to allow any leakage of steam. The pressure plate is held from moving endwise by dowel pins, or other device, which permit it to lift to afford a relief in case there is water in the cylinder. To return the plate to its seat, springs are provided on its back; and after it is once seated the steam pressure on the back holds it in place. The wear may be taken up by scraping the edges of the distance pieces. The pressure-plate is made very thick to prevent it from deflecting enough to clamp the valve, the deflection being caused by the steam pressure on the back of the plate. The valve takes steam at the ends and exhausts at the middle, it is auxiliary-ported to both the steam and exhaust, is perfectly balanced, practically frictionless and wearless, affords relief to water and, being of the "skeleton" type is of minimum weight.

Prof. Sweet added the rib R to protect the surface of the pressure-plate from any erosion which might be caused by the impact of the exhaust steam. This rib also strengthens and stiffens the end.*

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*The student is advised to construct a model of the Sweet valve similar to that shown in Fig. 47 for the Allen valve, using two pieces of stiff paper, one piece 3 inches x 7½ inches, and the other 1 inch x 9 inches. Only the head end of the valve need be considered. The guide straps on the larger piece of paper should be 5 inches apart, and the steam edge of the valve seat should be 1½ inches from the left strap. The end of the valve should be 3 in. from the left end of the slide. For dimensions (referring to Fig. 60) use W 1 inch, S ⅜ inch, E 1-16 inch, B ½ inch, a ¼ inch, C ¾ inch (purposely made short), D ½ inch, throw 1 inch and thickness of valve metal 5-32 inch.

(a) Referring to Fig. 55 and considering the valve as moving to the right, the **phases of opening** of the steam edge are as follows: (1) There is **double-ported** action (Fig 55a), caused by y and f opening at the same rate, until the opening at f becomes equal to that through the auxiliary passage (a). (2) With further movement (Fig. 55b), the opening at y continues to increase while that through (a) remains constant, i.e., the opening is **single-ported plus a constant.** This phase continues until (a) begins to be throttled by the exhaust edge of the valve seat. (3) With the movement continuing, (a) is throttled at the same rate as the main steam edge opens (Fig. 55c), the effective opening remaining **constant** until the auxiliary passage is entirely closed. The extent of the opening is equal to the width of port (W) minus the width of the bridge-end (B) or (W — B). (4) After the auxiliary port is entirely closed (Fig. 55d) the opening is of course **single-ported** until the port is wide open.

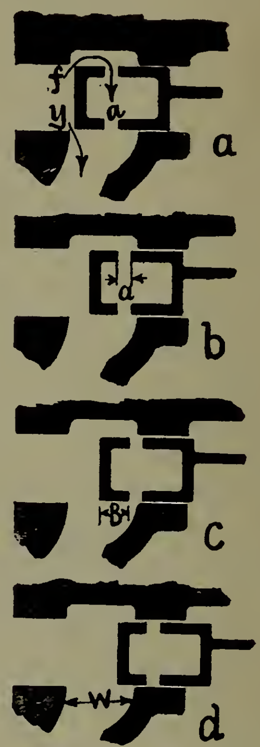


Fig 55

Now, if the valve moves to the left to close, the effective opening will decrease with these phases occurring in the reverse order. The opening to the exhaust may have some or all of these phases, depending on the proportions.

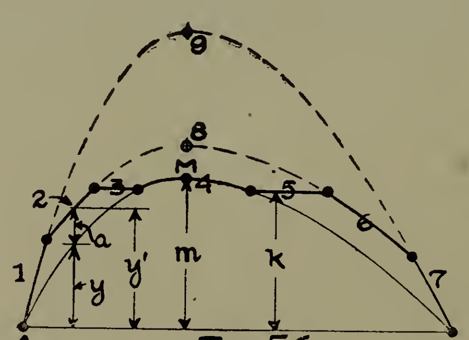


Fig 56

It is seen that the phases of opening are exactly the same as in the case of the Allen Valve.

(b). **A Diagram of Openings** of the steam edges of the valve is shown in Fig. 56, in which the ordinates y of the elliptic curve AMC represent the openings of the edge y in Fig. 55. The ordinates of the curve 1 — 2 — 3 — 4 etc. show the total effective opening, including that through the auxiliary passage. If y' is the ordinate of this curve, its value in the successive phases (numbered as in the preceding paragraph) is: (1) $y' = y+f = 2y$; (2) $y' = y + a$; (3) $y' = W-B =$ const.; and (4) $y' = y$. The maximum opening of the main steam edge is m.

(c). **Case 1.** If $W < B + m$, the case is that which has just been discussed.

(d) **Case II.** If $W = B + m$, lines 3 and 5 will be tangent to AMC at M.

(e). **Case III.** If $W = B + m + a$, curves 2 and 6 will meet at 8.

(f). **Case IV** If $W = B +$ (the desired opening w') and if $a = \frac{1}{2} w'$, the opening is double-ported throughout. Then for the steam edge the throw of the eccentric need only be one half that required for a single-ported valve. However, it may be necessary to use a larger throw to give sufficient opening for exhaust.

(g) With this valve, just as was the case with the Allen valve, the clearance volume is variable, that in the cylinder being augmented at times by that in the auxiliary passage in the valve. When a change takes place in the clearance volume the expansion line on the indicator card also changes, but with this valve the change is usually so slight as to be negligable.

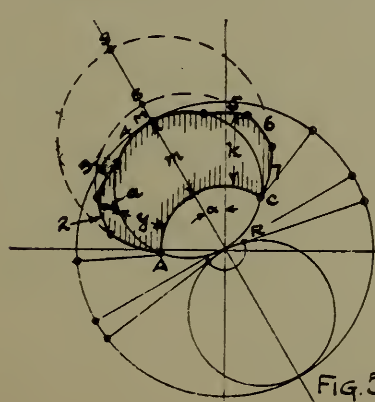


Fig. 57.

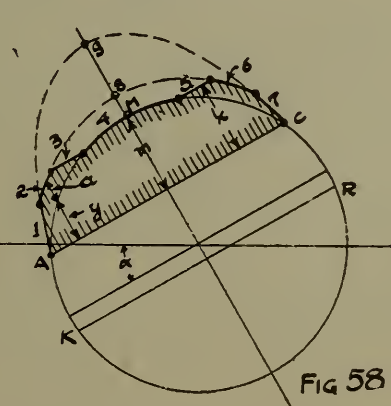


Fig 58

(h). The **Zeuner and the Sweet Diagrams**, for the steam edges of the Sweet valve, are shown in Figs. 57 and 58. In these

diagrams the figures and letters correspond with those used in the preceding paragraph in connection with Fig. 56. These diagrams also apply to the Allen Valve, the reference letters and figures being the same as those which were used in describing the diagram of openings for that valve (Paragraph 47b and Fig. 48).

(i) **Loss of Steam.** Considering the left end of the valve, if it moves far enough to the left so that the auxiliary-port travels beyond the end of the pressure-plate, this port fills with live steam and when the valve moves to the right again, this steam will be thrown over into the exhaust without doing work, because the exhaust edge is still open when the auxiliary-port opens to the cyl. passage. There are two ways of preventing this loss. One is to make the balance plate so long that the auxiliary passage will not overtravel. The other way is to make G wider than W in Fig. 60; then the live steam is transferred into the cylinder passage after the exhaust edge is closed, augments the compression and later is used during expansion. In this case, however, the exhaust is single-ported. To have double-ported exhaust and at the same time have G greater than W there may be employed an auxiliary port for the exhaust, made separate from that for the steam, as a′ in Fig. 59. This last arrangement is the original form of Sweet Valve.

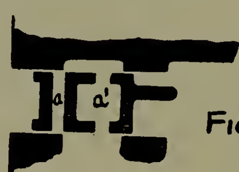


FIG 59

Considering this last form of valve—if, as it moves to the right, admission takes place before a′ is closed by the exhaust edge, this auxiliary passage will fill with live steam. Then, if the valve moves far enough to the right, so that a′ opens to the exhaust chamber, this steam will be lost without its having done any work. There are two ways of preventing this loss:—One is to make D (Fig. 60) so wide that a′ will not overtravel; the other is to so locate a′ that it will be closed by the exhaust edge before admission occurs.

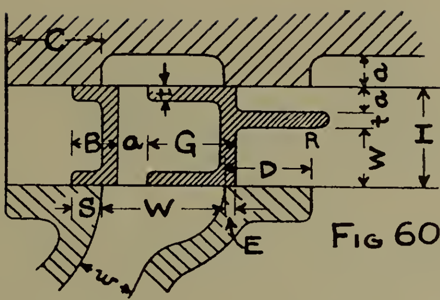


Fig 60

(j) **Proportions.** Referring to Fig. 60., B may be made any size that is consistent with strength and casting qualities.

w = opening to exhaust.

w′ = opening to steam (total).

m = max. opening of the main steam edge.

For the meanings of the other letters see the figure.

The height of pockets in the balance-plate and that of the space above the rib R must be at least equal to a. The height of rib R above the main valve-seat must be at least w. Hence the depth of the valve must be at least H = w + t + a.

If the auxiliary-port does not increase the maximum opening of the valve, but merely augments the initial and final openings (as in Cases I to II) the throw, lead and laps will be the same as those found for a single-ported valve. The amount of opening during the third phase (that of constant opening) may be arbitrarily assumed at any suitable value k; then W = k + B, but must, of course, be greater than w.

The width of the auxilliary-port (a) may be made anything within reasonable limits. It must be wide enough to permit the use of a core of thickness sufficient to withstand the flow of the molten metal when the casting is poured, and for this the width should usually be a little greater than the thickness of metal in the valve.

To receive the full benefit of the auxiliary-port during the whole of the period of opening, W must be at least equal to B+m+a, as in Case III.

To have double-ported action during the whole period, not only must W be equal to B + w′, but (a) must equal ½ w′. The throw would be that of a double-ported valve, provided it gives sufficient opening to exhaust, as in Case IV.

Provision must be made against carrying steam over into the exhaust, as explained in paragraph (i).

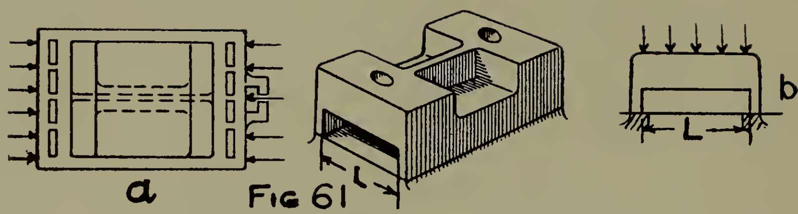


Fig 61

(k). **Strength.** The valve should be of the least weight consistent with strength, rigidity and casting qualities. The thickness of metal may be about one-third that of the cylinder wall (page 52), or t = (cylinder diam ÷ 60) + .1″

The valve is loaded on the ends by the steam pressure, as shown in Fig. 61a. The ends must be designed as beams. The addition of posts across the auxiliary-passage greatly strengthens and stiffens these ends, and if the valve is very wide one or more central bars should be added as shown dotted. The side strips which connect the two ends are compression members.

The **Pressure-Plate** must be made very thick, otherwise it will deflect and clamp the valve. It is a beam (Fig. 61b) of length L, uniformly loaded by the steam-pressure and supported at the sides. The deflection should be limited to about .001″ For such a beam the deflection is

$$\triangle = 5/384 \frac{p\,L^4}{E\,I}$$

Where p = load per inch of length (steam pressure).
L = length of beam, here a little greater than the length of the port.
E = 15 to 17 million for C.I.
$I = bh^3 \div 12$.
b = width of beam.
h = depth of beam.

A strip 1″ wide may be taken, (i.e. b = 1″).

Over the exhaust cavity the thickness may be made less than h, as the deflection at this place will not interfere with the valve.

Dowel pins, or other device, must be provided to keep the balance-plate in position and springs must be used to return the plate to its seat in case it should be lifted therefrom when there is water in the cylinder.

X206. Given, Cylinder diameter 16 in., stroke 18 in., length of port 16 in., C.O. ¾ stroke,

lead ⅛ in., compression 85% of stroke. Sweet valve, Case I.

(a) Compute the widths of valve openings for both the steam and the exhaust, using for the respective velocities of steam 8,000 and 5,500 ft. min.

(b) Draw a Bilgram Diagram as for a single-ported valve.

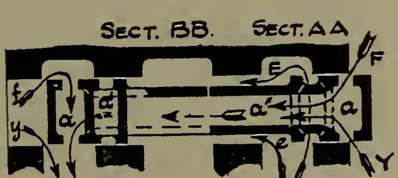
(c) Assume the width of opening (k) during the third (or constant-opening) phase, as ⅞ of the maximum steam opening found in (a). Take B the thickness of the metal at the end of the valve as ¾ in. Determine the width of port opening W.

(d) Assume the width of the auxiliary passage as ⅜ in. and draw a longitudinal section of the valve and balance plate. The thickness of this latter should be computed. Provide against loss of live steam.

(e) Draw a Diagram of Openings for the head end of the valve.

(f) Draw Zeuner and Sweet Diagrams.

Similar problems may be devised for the other cases.



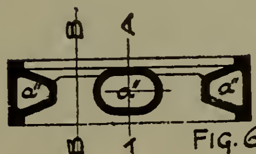


FIG. 62.

52. WOODBURY VALVE. A valve of this type is shown in Fig. 62. It combines the Sweet and the Allen principles. There are auxiliary passages a and a′, like those in the Sweet valve, but in addition there are also longitudinal ones a″, like that in the Allen valve. The opening to steam is quadruple-ported, the edge of the valve uncovering simultaneously at f, y, F. and Y. The opening to exhaust is double-ported.

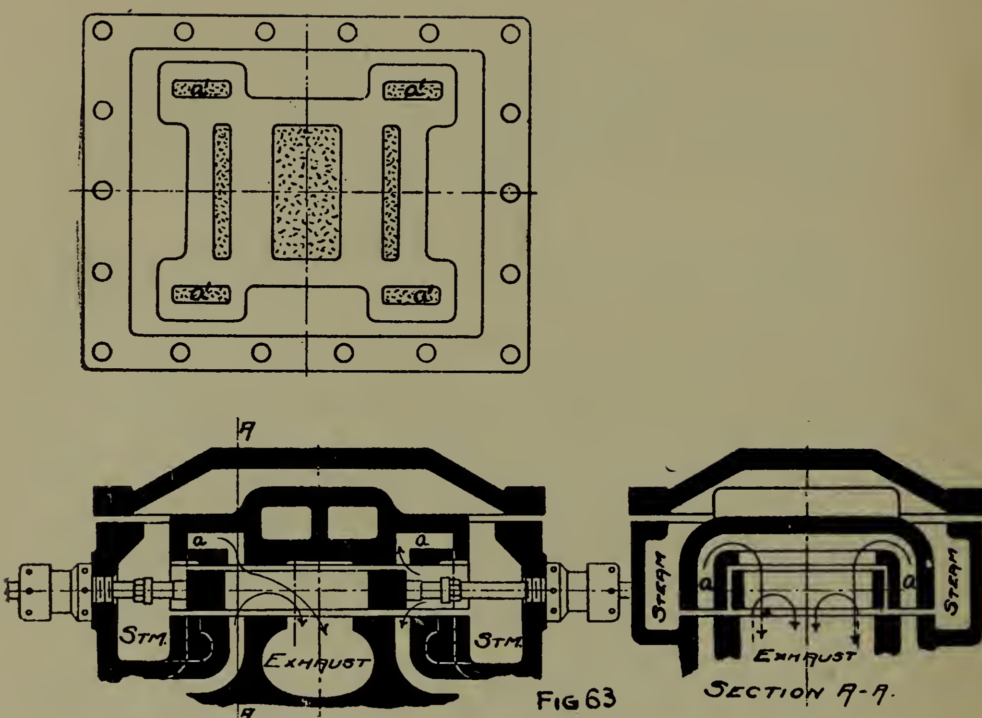


Fig 63

53. McEWEN (RIDGWAY) VALVE. Fig. 63. Instead of having the auxiliary passages in the valve itself, they are here located in the pressure-plate, as shown at (a) in the sectional views. The steam which passes the upper edge of the valve enters the passage (a), which connects with the auxiliary ports (a′) in the valve seat. These latter openings connect with the main passages in the cylinder. The valve gives double-ported opening to both the steam and the exhaust.

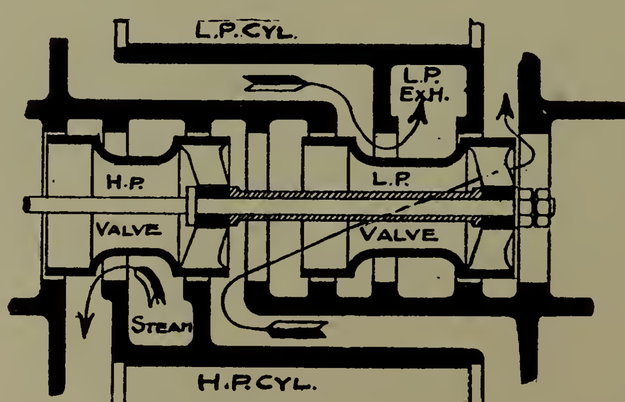


Fig. 64.

54. COMPOUND VALVES. Fig. 64 shows a pair of valves, which are mounted on the same valve stem and which are so arranged as to distribute steam to the H.P. and L.P. cylinders of a cross-compound engine, the cranks of which are 180 degrees apart. The valves are piston valves of the simple "D" type. That for the H. P. cylinder takes steam from its middle, while that for the L. P. cylinder admits steam past its ends.

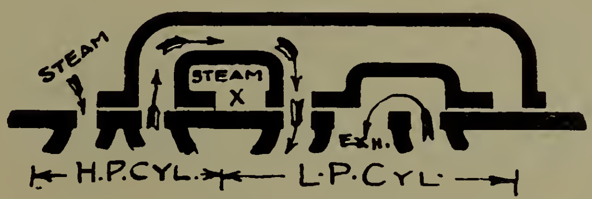


FIG 65.

Fig 65 shows a somewhat similar arrangement of a double-valve for a tandem-compound engine. There are simple "D" valves for both the H.P. and the L.P. cylinders. The walls of the exhaust chamber of the H.P. valve are extended to form a hood, or steam chest around the L.P. valve. The passage X is open at the side of the valve and is filled with live steam.

Piston valves similar to this last can be devised.

55. WESTINGHOUSE COMPOUND VALVE.

The Westinghouse compound engine is single acting, with cranks placed 180 degrees apart. One piston valve is used to distribute steam to both cylinders. When this valve is in the position shown in Fig. 66, the steam is being admitted to the H.P. cylinder through the port A, and exhausted from the L.P. cylinder at B.

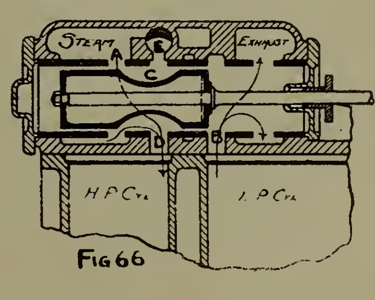


FIG 66

As the valve moves to the right, first, steam will be cut off at A, then the L.P. exhaust will be closed at B and finally C will establish communication between ports D and B, so that the H.P. cylinder will exhaust into the L.P. cylinder.

As the valve returns to the left, first, the L.P. cut-off and H.P. compression will occur simultaneously, then L.P. release and finally H.P. admission will take place.

E is a by-pass valve which is used in starting the engine.

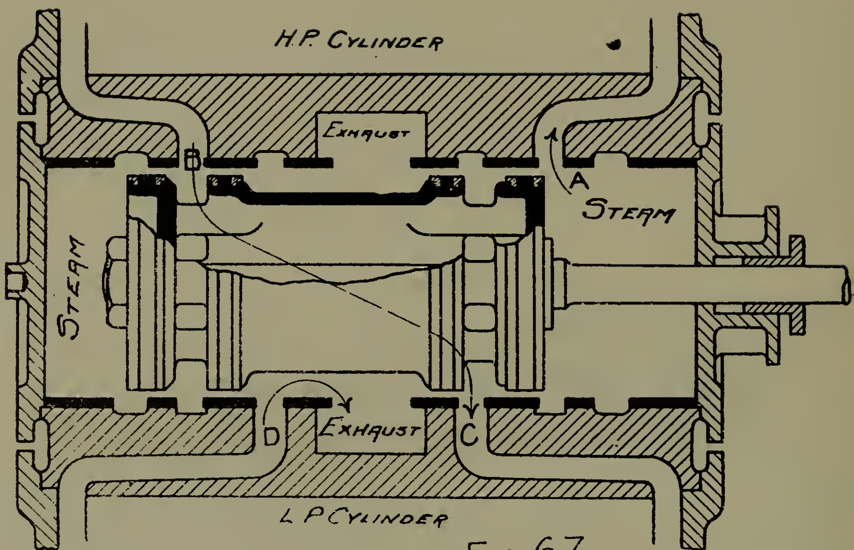


Fig 67

56. VAUCLAIN VALVE. Fig. 67. This valve is used on compound locomotives, on which the H.P. and L.P. piston rods are attached to the same cross head, so that the pistons move in unison. Only one valve is used to distribute the steam to both cylinders. In the Figure both pistons are moving to the left. Steam is being admitted to the H.P. cylinder at A, and exhausted therefrom at B into the middle of the valve. The right end of the L.P. cylinder is receiving steam through port C from the left end of the H.P. cylinder. The L.P. cylinder is exhausting at D.

The middle part of the valve is a simple piston valve for the L.P. cylinder. The two end portions constitute the valve for the H.P. cylinder. Both valves admit steam past their ends and exhaust at the middle.

OTHER FORMS. Valves of the pressure-plate type can be devised to take steam from the middle and to exhaust at the ends, but in this case the pressure-plate must be rigidly fastened to the cylinder, thus preventing relief from water.

In many cases piston valves can be made similar to the flat valves which have been described.

There are many other forms of valves, but they are mostly modifications of those already described.

CHAPTER III.

SHAFT GOVERNORS.

57. GENERAL. The shaft governor is so called because it is mounted on the main shaft of the engine, being placed either in the fly-wheel or in a governor case. It is a device which automatically regulates the cut-off so that the engine will supply just enough power to meet the demand, and, at the same time, maintain a nearly constant rotative speed for all loads within the capacity of the engine. As it works automatically, engines having such governors are frequently called "automatic engines."

In general, a shaft governor has one, or two, pivotted "weight-arms," the centrifugal force acting on which is balanced by one, or more, springs which are so adjusted that there is a different speed and a corresponding definite and distinct position of the arm, or arms, for each different load on the engine. The "weight arms" are connected either directly, or by links, to the eccentric, so that for each speed there is a definite and distinct position of the eccentric, a corresponding cut-off, and a definite amount of power developed. If the load changes, the speed of the engine will also change until a cutoff is found which gives the right amount of power to balance the load.

All so-called "constant-speed" governors are anomalies; for, while it is their function to maintain constant speeds—for them to act, it is necessary that a change in speed take place. However, if the governor is of good design and properly adjusted, the amount of variation is very small (being from 1 to 2½% of the "normal" or average speed), so that the speed is practically constant.

There are two general types of shaft governors—the "Centrifugal" and the "Inertia."

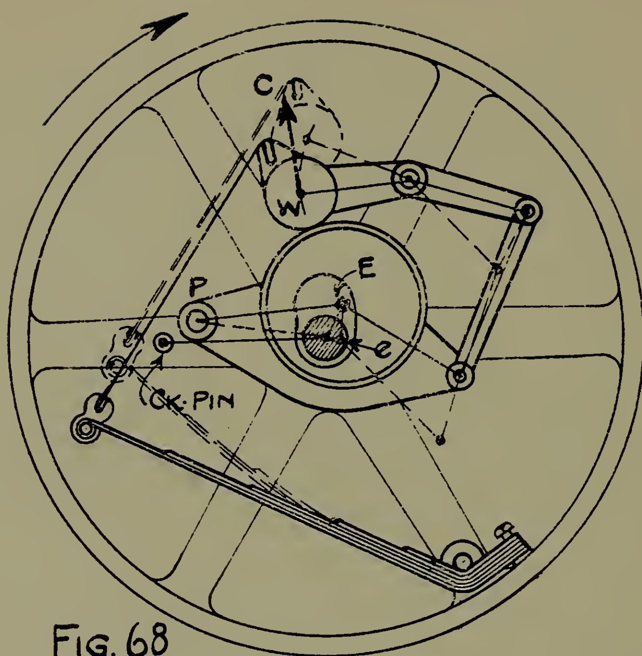


FIG. 68

58. CENTRIFUGAL GOVERNORS. The Sweet Governor, which was one of the earliest of this type, is shown in Fig. 68. Pivotted to one of the arms of the fly-wheel, or governor-wheel, is a "weight-arm," which has a heavy head W. When the engine is not running, this weight-arm is held in the "inner" position (that shown in full lines) by the leaf spring; and after steam is turned on, it will remain in this position until the speed has reached a certain point (for example, say 198 R.P.M.), when the centrifugal force C will just balance the spring pull. Now, if the speed is raised farther, the increased centrifugal force will cause the arm to move outward until, at some speed (say 202 R.P.M.) it reaches the extreme "outer" position (that shown by the dotted lines). At the "normal" speed (200 R.P.M.) the weight-arm would be about midway between these extreme positions; and for every other speed (between the 198 and 202 R.P.M.) there are definite positions of the arm.

In the example the total variation in speed is 2% of the normal R.P.M. By changing the adjustment of the spring, however, the amount of variation can be changed, but if it is made too small, the friction and inertia of the valve gear, and the other disturbances, will make the action of the governor uncertain, so there is a practical limit to the closeness of regulation.

Again referring to Fig. 68, it is seen that the eccentric is pivotted at P to one of the arms of the wheel, and is connected by a link to an extension of the weight-arm. When this latter is in the inner position, or is "in," the center of the eccentric is at E., the position for the latest cut-off; and when it is "out," the eccentric-center is at e, the position for zero cut-off.

The manner in which the governor operates is as follows:— When the engine is standing still, the governor holds the eccentric in the position E for the latest cut-off. Then, if steam is turned on, the engine will speed up until a certain R.P.M. is reached at which the governor-arm will begin to move out, thus shifting the eccentric towards e and decreasing the cut-off. This movement continues until a position is reached at which the power developed just equals the load, and as long as this latter remains constant, the governor-arm will remain in this position. Now, if the load is reduced, the engine will speed up (tending to run-away) and this causes the weight-arm to fly out, shifting the eccentric towards e and reducing the power developed until it becomes again equal to the load. Similarly, if the load is increased, the speed of the engine will decrease, and, as the weight-arm moves "in," the cut-off will be increased, until at some position of the arm a balance is again reached between the power and the load.

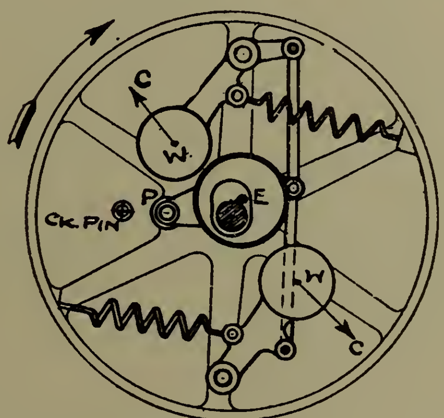


Fig. 69.

Fig 69 shows another "Centrifugal" Shaft Governor; but in this case there are two weight-arms, symmetrically placed, instead of one. In its action, this governor is identical with that which has just been described.

In both Figs. 68 and 69, it is seen that the eccentric has a hole in its central web, made large enough to clear the shaft as the eccentric swings from one extreme to the other. In both cases the pivot is placed on the same side of the shaft as the crank pin. It is possible, however, to arrange the governor to have a pivot on the opposite side, but this does not give as good a path for the center of the eccentric as does the former arrangement. The direction of rotation is such that the weight-arms trail behind their fulcrum-pins or pivots.

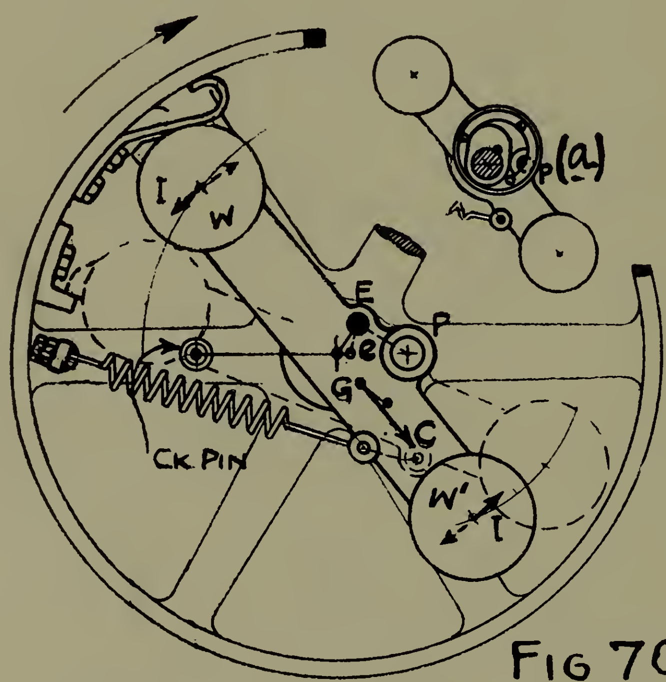


FIG 70

59. INERTIA GOVERNORS.—Fig. 70 shows the Rites Inertia Governor, which consists of a long weight-arm (WW′), an eccentric pin E, and a spring. The arm is pivotted at P, close to the shaft, and its end W′ is heavier than W, so the center of gravity is at G. The position of the parts shown in full lines is for latest cut-off, and is the one occupied when the engine is not running; that shown by the broken lines, is for the earliest cut-off. In the former position, the arm is said to be "in," and in the latter, "out." The direction of rotation is shown by the arrow.

As the engine starts up, the governor-arm remains in the inner position until a certain speed is reached, when the centrifugal force C, acting on the weight-arm, becomes great enough to bal-

ance the spring pull. Then, with a further increase in speed, the weight arm will move out, (the eccentric meanwhile moving toward e) until a sufficiently early cut-off is obtained.

Now, if the load falls off, the engine will speed up, and the increased centrifugal force will cause the weight-arm to move out until the cut-off is reduced to the proper amount, the action being just the same as in the case of the Centrifugal Governor. However, in addition to the centrifugal force acting on the arm, there is also an inertia force which assists the movement.

The inertia acts in this manner:—As the engine speeds up, the governor-arm tends to continue to rotate at its old speed, because of its inertia, and hence lags behind the wheel, moving with respect to the latter in the directon shown by the arrows I-I in the figure. It is seen that this movement is in the same direction as that caused by the centrifugal force. Again, if the load is suddenly increased, the engine will slow down, but, because of its inertia, the weight-arm will continue at its old speed, thus gaining on the fly-wheel, and again assisting the centrifugal force in causing the motion.

It is seen that the Inertia Governor is primarily a Centrifugal Governor, but that, in addition, the weight-arm is so pivotted, and has its weight so distributed that its inertia helps to cause the adjustment to take place, and that the more sudden the change in the load, the greater will be the assistance rendered.

The eccentric or eccentric-pin is usually mounted directly on the weight-arm, or, in the case of the eccentric, is keyed directly to the end of the fulcrum pin opposite to that to which the arm is fastened. With these arrangements, in order to have the inertia of the weight-arm act in the right direction, **the fulcrum pin must be placed on the side of the shaft opposite to the crank pin,** when an external valve is used; and on the same side, when the valve is internal.

On center-crank engines the governor is frequently placed in the outer side of the wheel, in which case, since the shaft does not extend beyond the governor wheel, the arrangement can be that shown in Fig. 70. If, however, the governor is placed on the side of the wheel next to the engine frame, both the governor-arm and the eccentric must be made to surround the shaft in the manner shown at (a) Fig. 70.

60. SUMMARY. For both forms of shaft governors, it has been seen:—

(a) That there is a definite speed, cut-off, and power for each position of the weight-arm.

(b) That when the arm is "in," the speed is the lowest and the cut-off is the latest; whereas, if the weight-arm is "out," the reverse is the case.

(c) That an increase in load decreases the speed and causes the arm to move "in," which gives a later cut-off; whereas, the effect of a decrease in load is the reverse.

(d) That for close regulation, the friction and inertia of the valve-gear parts must be small, and especially is this necessary when the inertia form of governor is used.

(e) In addition, in the case of the Inertia Governor, the fulcrum-pin must be placed opposite to the crank-pin if the eccentric is attached directly to the governor-arm (or to its fulcrum-pin), and if an external valve is used.

There are almost an unlimited number of forms of shaft governors, but all of them are merely modifications of those which have been described.

CHAPTER IV.

GEARS WITH SINGLE VALVE AND VARIABLE ECCENTRIC.

61. GENERAL. In the chapter on shaft governors, it was seen that in the process of regulating the engine, the governor changes the position of the eccentric with respect to the crank, varying both the throw and the angle of advance. How these changes affect the action of the valve will now be investigated.

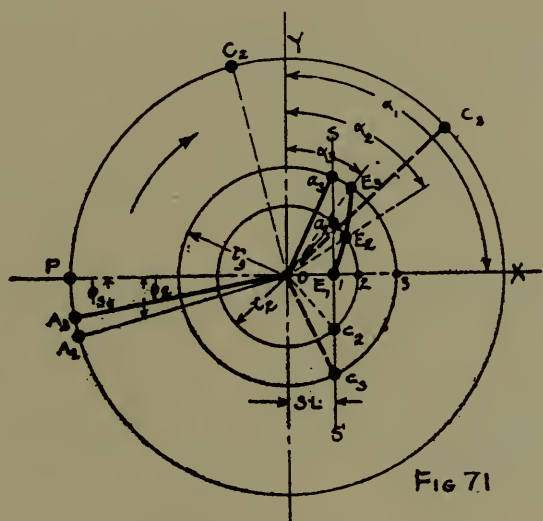


FIG 71

Fig. 71 is a "Diagram of Positions," showing the relative locations of the crank and the eccentric. The line SS′ is the "steam lap line," its distance from the Y-axis being equal to the steam lap. First suppose the angle of advance and the throw of the eccentric are respectively α_3 and r_3. Then, when the crank is on the dead center P, the eccentric will be at E_3. If now, the eccentric is rotated back to a_3, on SS′, it will be in the position for admission. The corresponding position of the crank is A_3. Starting from this phase and rotating the mechanism in the direction shown by the arrow, the valve will be open until the eccentric reaches c_3, when cutoff will occur. The crank position for cutoff is C_3 and it will be noticed that this event comes quite late in the stroke. The maximum opening of the valve is shown by the distance 1–3 ; the lead opening is equal to the distance that E_3 is to the right of the line SS′, and the lead angle is φ_3.

Next, suppose that the eccentric is moved to a new position with respect to the crank, so that it will be at E_2, when the crank pin is at P, the new angle of advance α_2 being greater than that in the previous case, and the throw r_2 being less. It will of course be found that this shifting of the

eccentric has changed the action of the valve. As the mechanism rotates in the direction of the arrow, admission takes place when the eccentric and crank are respectfully at a_2 and A_2 ; and cutoff occurs with these at c_2 and C_2. It is seen that cutoff is now much earlier than it was in the previous case and it is evident that this change was brought about by increasing the angle of advance. The maximum opening is now given by the distance 1-2, which is less than that in the previous case, as would be expected with the decreased throw. The lead is the distance between E_2 and SS′.

Again, suppose the eccentric is moved to a position E_1, just opposite the the crank P (the angle of advance being 90 deg.) with the throw just equal to the steam lap. Then the valve will not open at all but when the crank is on dead center it will have its maximum displacement and its edge will be just even with the port edge. Nominally the cutoff, the lead, the lead angle and the maximum opening are all zero.

From the forgoing it is evident that the cutoff and the lead angle (or lead) can be changed in any way that is desired by merely shifting the eccentric to the proper position with respect to the crank, and that the shifting also affects the width of valve opening. Later it will be seen that this also changes the exhaust events.

In Fig. 76 are shown theoretical indicator cards for several different cutoffs, and it is seen that when the cutoff is made to occur earlier, the release and compression also take place sooner.

The shaft governor, which controls the positions of the eccentric, should be so arranged that as the weight arms move "out," as they do when the load is decreased, the eccentric will be shifted "in" (toward E_1) to give an earlier cutoff, and vice versa.

62. THE BILGRAM DIAGRAM.

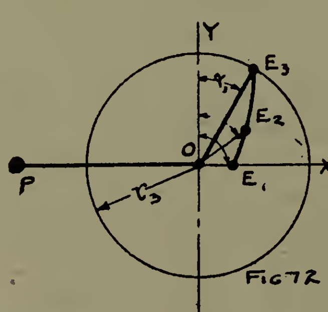


Fig 72

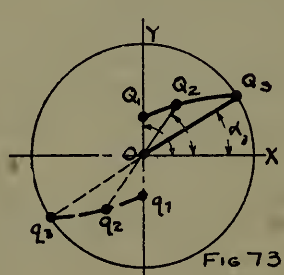


Fig 73

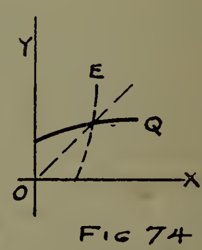


Fig 74

In Fig. 72 is shown the eccentric-path E_3E_1, the crank being on the dead center P. When the eccentric is at E_3, the throw is OE_3 and the angle of advance α_3. On the Bilgram diagram, Fig. 73, the position of the lap circle center Q_3, corresponding to E_3, is found in the usual manner, by drawing OQ_3 at the angle α_3 with the X-axis and making OQ_3 equal to OE_3. The action of the head end of the valve, for this particular throw and angle of advance of the eccentric, may be studied by drawing the steam and ex-

haust lap circles with Q_3 as center, just as in the case of the ordinary gear with the fixed eccentric. The lap circles for the crank end would have center at q_3, diametrically opposite Q_3.

If the eccentric is now moved to a new position E_2, Fig. 72, it has a new throw and a new angle of advance. Laying these off on the Bilgram Diagram, Fig. 73, the new lap circle center Q_2 is located, and with this as center the lap circles may again be drawn and the action of the valve determined for this case.

The positions of Q, corresponding to the other positions of E, may be found in a similar manner. The curve through these points will be termed "the path of the lap circle center," or more briefly **"the lap-circle path."**

If Fig. 72 is superimposed on Fig. 73, as in Fig. 74, it is apparant that the **paths of Q and E are symmetrical with respect to the 45 deg. line bisecting the quadrant XOY.** It is also evident that, in Fig. 73, the paths of Q and of q are symmetrical with respect to O.

The shifting of the eccentric of course in no way changes the proportions of the valve, so for all positions of Q the lap circles would be the same.

Fig. 75 is a Bilgram Diagram for the head end of the valve, for three positions of Q. It shows that, **as the eccentric is moved "in" (i. e. as the lap-circle center is moved towards Q_1), all the events are made to occur earlier in the stroke,** (with the possible exception of admission,) **and that the maximum opening w is decreased.**

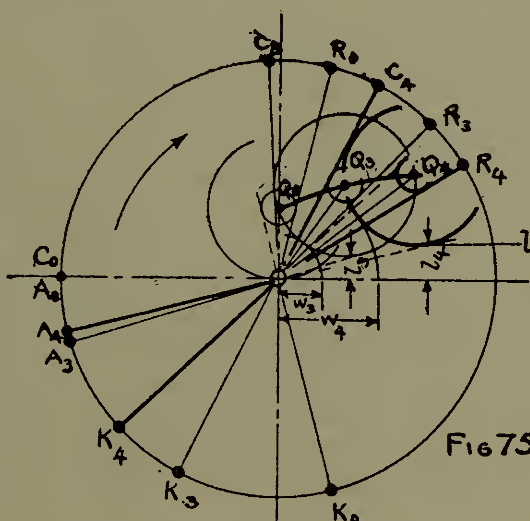


Fig 75

It further shows that the lead is decreased in this particular case. In general, however, the way the lead and lead angle (admission) vary depends on the character of the path of Q (and of E) ; and this to a certain extent, also has an influence on the widths of valve opening. So in designing, the character of the lap-circle path must carefully be considered.

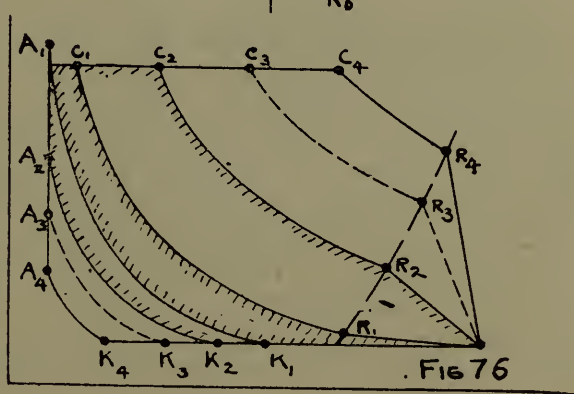


Fig 76

In Fig. 76 are shown theoretical indicator cards corresponding to two positions of Q shown in Fig. 75, and also for two other positions of this point, which are omitted from the former figure to avoid confusion. These cards show clearly how the release and compression vary with the cutoff.

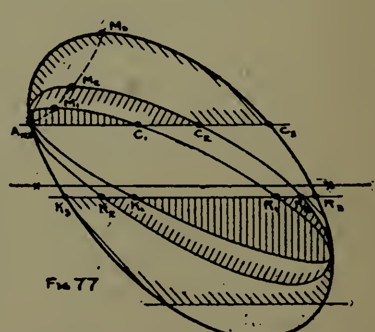
Suppose it is desired to **design** a valve gear of this type, and that latest cutoff is to occur when the crank is at C_4, in Fig. 75, with w_4 as the maximum opening of the valve, and l_4 the lead. On the Bilgram Diagram, just as for an ordinary slide valve, there would be drawn the lead line l, the arc with radius equal to the desired maximum opening w_4, and the crank position OC_4 for C. O.; then Q_4 would be so located that the lap circle would be tangent to all three of these lines, thus determining the steam lap, the throw and the angle of advance. Then assuming one of the exhaust events, the size of the exhaust lap would be fixed and the proportions of the valve and eccentric would be determined.

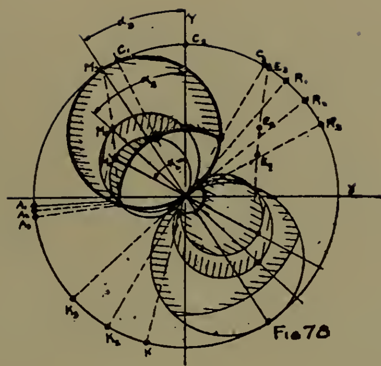
It is next necessary to decide on the character of the path of Q (and of E). In this case, this path is an arc of a circle drawn in such a way that when the lap circle center is at Q_0 the lead and cutoff are zero.

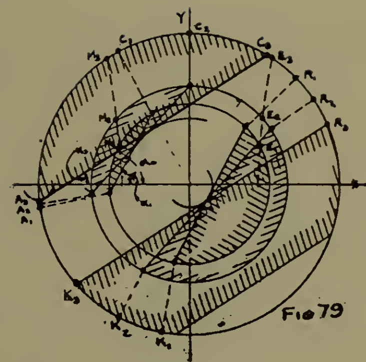
Having drawn the path of Q, suppose it is desired to study the action of the valve when the cutoff occurs with the crank at some position, say C_3. To determine the location of the lap circle center, we know that it must be on the path of Q, and that the circle itself must be tangent to OC_3; thus its position is definitely fixed. The Bilgram Diagram for this position can then be completed, and the action of the valve be studied.

In determining the **size of the exhaust lap** it is necessary to consider both the minimum and the maximum amounts of compression that are obtained. When the cutoff is that for friction load it is desirable that the pressure at the end of compression should not be above boiler pressure; for if it exceeds this, the valve will clatter, if of the lifting type, and if not of this type an undesirable excess in pressure results. On the other hand, when the cutoff is latest, the compression is least and it should then be great enough to properly cushion the reciprocating parts. The range of compression should be kept within these limits if possible.

63. ELLIPTICAL DIAGRAM. Fig. 77. For each position of the eccentric in its path, there will be a separate ellipse, which can be obtained in the usual way. The diagram shows clearly how the shifting of the eccentric effects the valve-openings and events. The relative sharpness of these latter is indicated by the slope of the ellipses where they cross the lap lines. Thru the maximum points M there has been drawn a curve ("the locus of M") which will be referred to later.







64. ZEUNER AND SWEET DIAGRAMS. These are shown respectively in Figs. 78 and 79, and are constructed in the usual manner, taking each position of the eccentric in turn. It is seen that the loci of M and of E are symmetrical with respect to the Y–axis.

65. BARR DIAGRAM. In Fig. 80 (a) shows the path of a swinging eccentric when the crank is at P, and (b) is the corresponding Bilgram Diagram. For each of the three positions of E and Q shown, the crank-position M for maximum opening has been found, by drawing it at right angles to OQ. The corresponding positions of the piston are 1, 2 and 3.

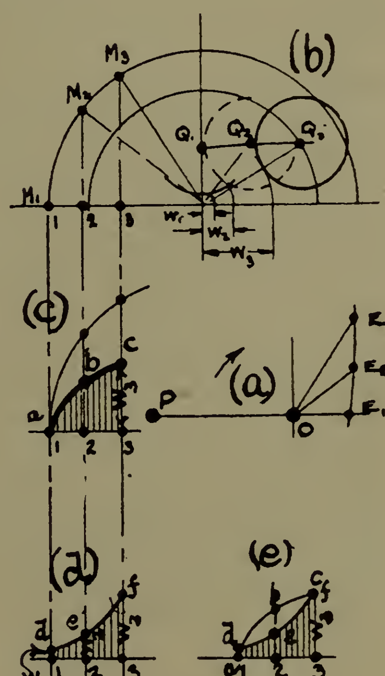


FIG 80

If the piston is assumed to have simple harmonic motion, its velocity, V, is proportional to the sine of the crank angle, or, what is the same thing, to the ordinate of the crank pin. Referring to equation 3 on page 42, it is seen that, for a given velocity of steam (v) and certain area of piston (A), the valve opening (a) should be proportional to the velocity of the piston (V), which we have just seen is a function of the ordinate of the crank pin. Therefore, in drawing a curve of desired openings, its ordinates would be made directly proportional to those of the crank pin. This curve, a–b–c in Fig. 80 (c), is of course an ellipse, which can be readily constructed as soon as one point has been determined.

Now suppose the desired width of opening for the latest cutoff (ecc. at E_3 in its path) has been found to be w_3. This maximum opening

occurs when the piston is at 3, at which point, in Fig. 80(c), w_3 would be erected as an ordinate, thus giving one point (c) on the curve. Through (c) the **curve of desired maximum openings** can then be drawn by constructing the elliptic arc a–b–c. Now, if the eccentric is moved to position E_2 in its path, the maximum opening then occurs when the piston is at 2, and from the curve just drawn it is seen that the width of opening should be equal to the distance 2–b, in order to have the same velocity of steam as before; and similarly for other positions of the eccentric in its path.

Going back to the Bilgram Diagram, Fig. 80 (b), and drawing the steam lap circles, the **actual maximum openings**, w_1, w_2 and w_3 are found. In Fig. 80 (d) these are shown plotted on their respective piston positions, giving the curve d–e–f, (which is seen to be the same as the locus of M in Fig. 77, when referred to the steam-lap line as a base.)

The curves of desired and actual, maximum openings should of course be identical, but in Fig. 80 (e) where they are both plotted on the same base, it is seen that the actual openings are smaller than the desired, except at the extremes. This reveals one of the faults of this type of gear, but it can be partly remedied in several ways.

One of the **ways of increasing the maximum openings**, for the intermediate cutoffs, is to give the path Q greater curvature in an upward direction; for in Fig. 80 (b) it is seen that if Q_2 is raised, the opening w_2 will be correspondingly increased. Another way is to increase the size and travel of the valve, the valve diagram remaining the same, except as to scale; but as this gives openings which are larger than are needed for later cutoffs, and especially as it involves increasing the size of the valve-gear parts. it is an undesirable method. A better way is to make the valve multiported, thus obtaining even larger openings than in this last case, without changing the eccentric. The maximum openings for a double-ported valve are shown by curve 6 in Fig. 81, in which 5 gives the openings for a single-ported valve and 7 those that are desired. It is seen, however, that for the later cutoffs the openings are still wider than are needed. But if the valve has an auxiliary port like that in the Sweet type of valve, the wider openings can be made to approximate quite closely those that are desired. The way this can be done will now we considered.

Let the positions of the **Sweet Valve** shown in Fig. 82, be the extreme ones corresponding to different locations of the eccentric in its path. Then the respective **phases of maximum opening** are seen to be the following:—(**1**) **Double-ported,** so long as the maximum displacement of the valve is such that the opening y, in Fig. 82a, is less than the width of the auxiliary port (a). (**2**) **Single-ported plus a constant,** (Fig. 82b), when the throw is greater than this amount, the constant quantity being equal to (a). (**3**). **Constant** (Fig. 82c) when auxiliary port is partly closed by the exhaust edge of the valve seat, the opening k being equal to (W–B). And (**4**) **Single-ported** (Fig. 82d) when the auxiliary port is entirely closed when the valve is in the extreme position.

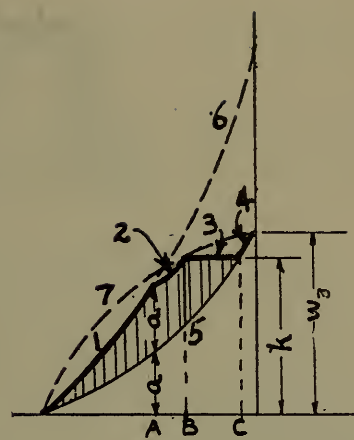


Fig 81

In Fig. 81 the maximum openings, for the different positions of the eccentric in its path, are shown by the heavy line 1–2–3–4, the parts being numbered to correspond with the phases just described. The first phase ends when the ordinate is equal to 2a and the second terminates when the ordinate equals k, which in turn is equal to (W–B).

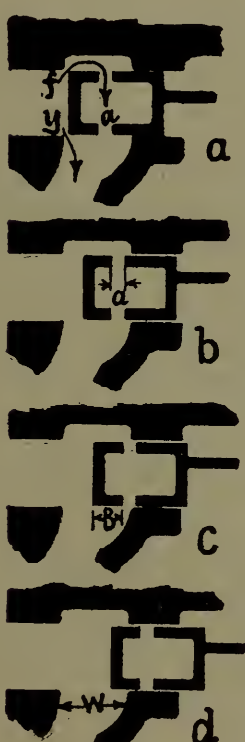


Fig. 82.

In **proportioning a Sweet Valve** for this type of gear, the curve of desired maximum openings (7) and that of the actual openings (5) for a single ported valve, would be drawn first. By doubling the ordinates of 5 the curve 1 for the double ported phase would be obtained. Then assuming a suitable width of the auxiliary port (a) curve 2 for the second phase can be obtained by adding the constant quantity (a) to the ordinates of 5. The height (k) of the line 3 for the third phase can be so drawn as to make the curve approach 7 as close as possible. Then assuming the width B of the bridge-end of the valve, the width of port W in the valve seat is fixed (since $W = k + B$) and this must of course be at least as large as the width of the cylinder passage. The valve and valve seat can then be drawn, using the values of a, B and W which have been determined. If the valve is made quadruple ported, as in the case of the Woodbury (§ 52), a still closer approximation to the desired openings can be obtained, but this is at the expense of greater complication.

The above method of determining the proportions of the Sweet Valve is due to Professor J. H. Barr.

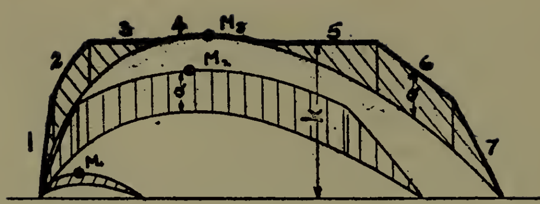


Fig 83

Fig. 83 shows a **Diagram of Openings**, which is that part of the Elliptical Diagram which lies above the steam lap line. This has been corrected to show the effect of the auxiliary port in the valve. The diagram for the greatest travel is similar to that shown in Fig. 56 and shows all four phases of opening and closing. The intermediate curve shows only two phases, and the smaller one, one phase. This diagram has been drawn very much distorted for the sake of clearness. If it is drawn accurately it will be found that on it the locus of M is the same curve as 1-2-3-4 in Fig. 81.

On the **Zeuner and the Sweet diagrams** in Figs. 78 and 79, the effect of the auxiliary port can also be shown, as it is in Figs. 57 and 58, for the greatest travel.

66. THE PATH OF THE ECCENTRIC. In selecting this, one must consider not only the manner in which the eccentric is to be moved by the governor, but also how the character of the path influences the action of the valve.

In general there are two different ways in which the eccentric is moved. In the first, it is shifted over a straight-line path, and is called a "**shifting eccentric.**" In the second it is swung about a pivot, and is therefore called a "**swinging eccentric.**" This latter, arrangements of which are shown in Figs. 68, 69 and 70, is the one more commonly adopted as it usually involves less complication in the governing mechanism than does the former.

The character of the path of the eccentric has an important influence on the steam distribution, as it affects the lead angle, the exhaust events, and the valve opening.

First we will consider the "**shifting eccentric**", taking up two general cases.

Case I. Constant Lead-Opening. In Fig. 84 at (a) is shown the eccentric-path, which is parallel to the Y-axis, and gives constant lead. It is seen in the corresponding Bilgram Diagram, (b) in this same figure, that as Q is shifted to the left, the admission, A, is made to occur earlier; that altho the lead opening remains constant as shown by the line l, the lead angle ϕ varies greatly, which ordinarily is undesirable.

Further it is evident that it is not possible to obtain a cutoff at zero stroke, for no matter how far Q is shifted to the left, the crank position OC for cut-off can never become horizontal. However, it is not usually necessary to

provide for as early a cutoff as this, since there is always the friction of the engine to be overcome. Further, the very early admission assists the cut-off in reducing the area of the indicator card.

Ordinarily the position of the eccentric below E_0 in Fig. 84a, would be that for rotation in the opposite direction; but in this case where the lead-opening is large, if the engine is once started, the direction will not be reversed, until E is shifted below the point E_1 some little distance.

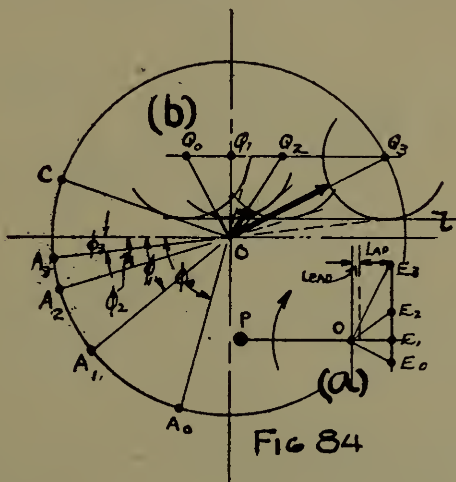


Fig 84

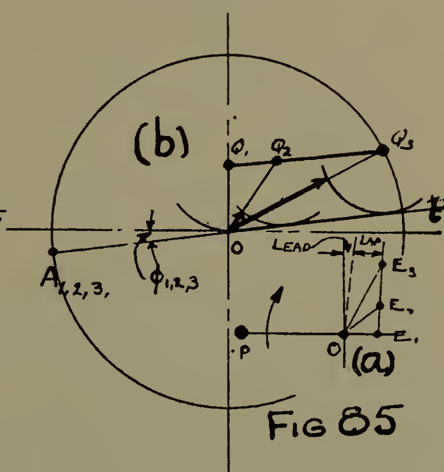


Fig 85

Case II. Constant Lead-Angle. In Fig. 85a it is seen that as the eccentric is shifted "in" the lead is decreased uniformly, becoming zero when E_1 is reached. The Bilgram Diagram shows that the crank position A for admission remains the same for all positions of Q, the lead-angle being constant. It is also seen that zero cutoff can be obtained. Ordinarily these are desirable attributes.

In both Figs. 84 and 85 the maximum width of valve opening is the same when the lap circle center is at Q_3. But, when Q is in the other positions, it is seen that in Fig. 84 there are wider openings than in Fig. 85, and Case I would therefore be the one selected if there was nothing else to be considered beside the openings.

In Fig. 86, in which are shown both cases, OC is the crank position for the normal cutoff (say at ¼ stroke). QI and QII are the respective lap circle centers for the two cases, and are of course at a distance from OC equal to the steam-lap. For convenience suppose the exhaust lap circles are of zero radius. It is seen that both **release and compression** occur later in Case II than in Case I, and as these events usually come earlier than is desired, Case II would be selected if there were no other considerations involved.

In the case of the **swinging-eccentric**, the path, which does not usually have very great curvature, may be considered as approximating the straight line path of the shifting eccentric, so that all that has been said in regard to this latter also holds for the swinging eccentric with slight modification which will now be considered.

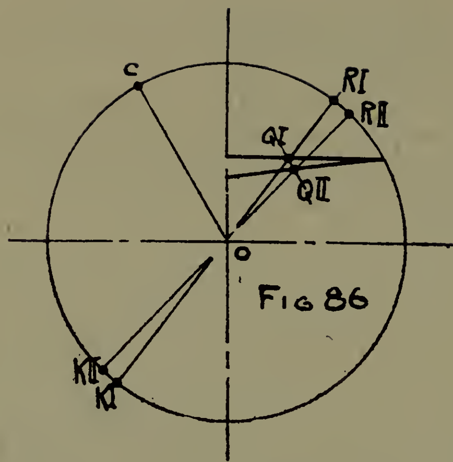


Fig 86

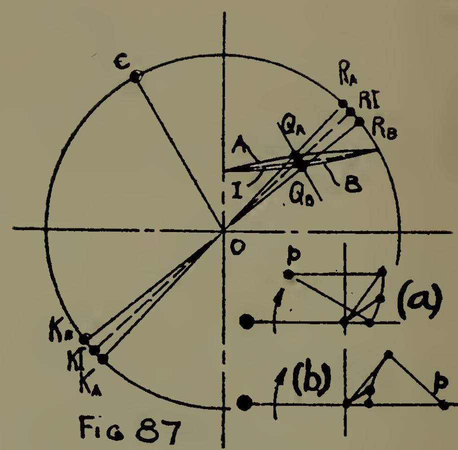


Fig 87

Case A. In Fig. 87, (a) shows an arrangement of the swinging eccentric like that shown in Fig. 68, the pivot for the eccentric being located at p, on the same side of the shaft as the crank pin. In the Bilgram Diagram A is the corresponding path of Q.

Case B. In Fig. 87, (b) shows an arrangement like that ordinarily used in connection with the inertia type of governor illustrated in Fig. 70, the eccentric-pivot, p, being placed opposite the crank. On the Bilgram Diagram, B is the corresponding path of Q.

Comparing the last two cases with each other, it is seen that Case A gives the greater openings, while the exhaust events occur later in Case B. It is also seen that the straight line path of Q (I in Fig. 87) gives openings and exhaust events intermediate between these two cases.

From the foregoing discussion, it is evident that the selection of the eccentric path is a matter of compromise. As to which kind of path is the best, opinions differ, and no general rule can be given. One must decide what things are most important for the particular case at hand, and favor those as much as is allowable. In selecting the path the following **general rule** will be of assistance :—For any given cutoff, a change in the path involving the raising of Q, will increase the valve-opening, the lead, the lead angle, the release and the compression ; whereas a change in the reverse direction will of course have the opposite effect.

Since the admission and the compression are, in a way, complimentary to each other, the admission can occur quite late, when the cutoff is very early, since in this case the compression pressure is very high. Some designers even use negative lead for the very early cutoffs.

It is of course desirable to be able to use the same governor wheel and parts, no matter which way the engine runs, and in selecting the path this should be kept in mind. When the governor is of the Rites inertia type, the eccentric pivot is usually placed on the center line of the crank for this reason.

67. EQUALIZATION OF EVENTS. For any one position of the eccentric in its path, the events can of course be equalized in the ways that were explained for the simple gear with fixed eccentric (§§ 22–24). Equalization for all positions of the eccentric in its path is impossible, but can be approximated more or less closely, by methods which will now be explained.

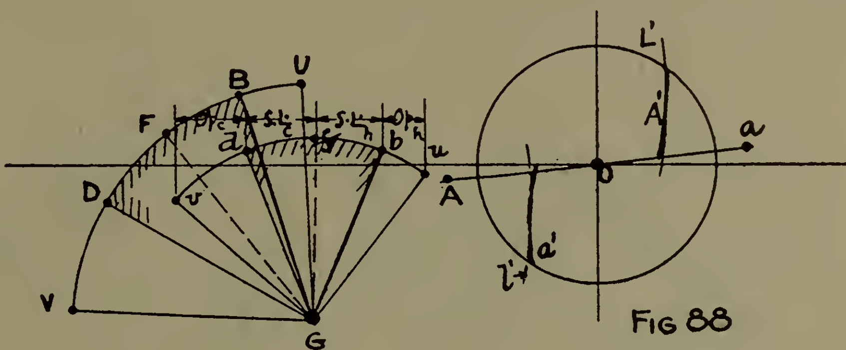


FIG 88

(a) Equalization of Admission. Supposing that it is desired to have constant and equal admission at all times in the two strokes, let A and a in Fig. 88 be the respective desired crank positions for these events and let A′ and a′ be the corresponding positions of the eccentric-path. Let BGb be the position of the rocker for admission at the head end, at which time the edge of the valve is of course even with that of the port; and let DGd be the corresponding rocker position for the crank end.

To have admission at the head end occur only when the crank is at A it is evident that all points on the path A′ must be at a distance from B equal to the length of the eccentric rod, or in other words, that an arc L′, struck with B as center and with a radius equal to the length of the eccentric rod must coincide with A′. Or, given the path A′, it is of course possible to find a position of B which will give coincidence between these arcs.

To have constant admission at the crank end the point D should be so located that the eccentric-rod arc l′ (struck with D as center) will coincide with the path a′, but this is not possible. Hence while constant admission can be obtained at one end of the cylinder, it can not be at the other. Since it is not possible to have the admission vary the same way at both ends, equalization of this event for all positions of the eccentric in its path is not possible in this case.

In general exact equalization of admission is obtainable only when the path of the eccentric is a straight line and the eccentric rod is of infinite length, so that, in practice exact equalization is never obtainable. However, it can be approached more or less closely by using as small a curvature for the eccentric path, and as long an eccentric rod, as the other considerations will permit.

Since an oblique guide is used, the valve motion is not harmonic, and hence the valve diagrams can be used only to get the throws and angles of

advance of the eccentric together with the crank positions for the events. The actual laps and openings can be obtained only after the positions of the rocker (or oblique guide) have been found as in Fig. 88. In this figure FGf is the central position of the rocker, UGu is its extreme position to the right and VGv that to the left. The steam lap for the head end is given nominally by the horizontal projection of the distance fb, while that for the crank end is the projection of fd. The projections of bu and dv give the corresponding maximum openings.

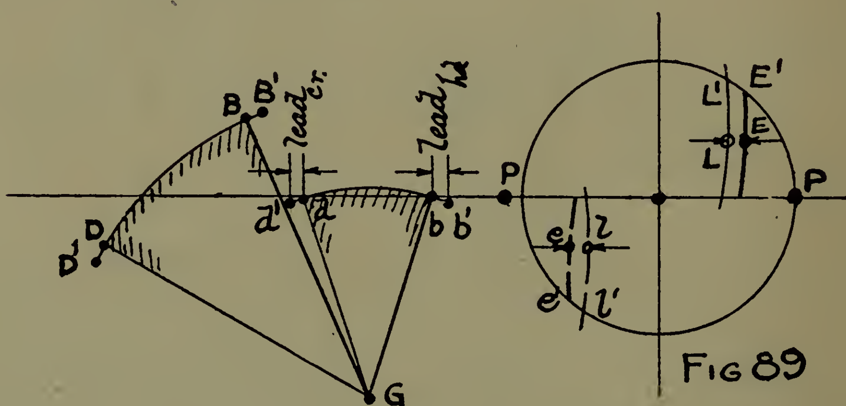


Fig 89

(b) **Equalization of Lead.** In Fig. 89, when the crank is on the head end dead center P (at which time the valve is open to lead) let E be one position of the eccentric in its path E′, and let B′Gb′ be the corresponding position of the rocker. As before, BGb is the rocker position for admission, and L′ is the eccentric-rod arc struck with B as center.

The lead opening, which is equal to the horizontal projection of bb′, can be produced by moving the eccentric through the horizontal distance LE, and it is evident that, no matter where the eccentric is in its path E′, its horizontal distance from the arc L′ is a measure of the lead, (but not necessarily equal to it).

Evidently the amount of lead, and the way it varies, can be controlled largely by the position selected for B. In designing, then, we would locate E, the position of the eccentric in its path for normal load, and lay off the distance LE to represent the lead; then we would so locate B that the distance between the arcs L′ and E′ varies in the way we wish the lead to change, as the eccentric is shifted over its path.

Similarly for the crank end, e′ is the path of the eccentric when the crank is on the dead center P′; e is the position of the eccentric corresponding to E; D′Gd′ is the rocker position for lead and DGd that for admission; l′ is the eccentric-rod arc struck with D as center; and the distance el is a measure of the lead for this position of the eccentric. It is desired to so locate D that, as the eccentric is shifted, the distance between l′ and e′ will vary in the same way as that between L′ and E′, but it is evident that this is im-

possible, so equalization of lead for all positions of the eccentric in its path is not possible in this case. As before, the desired result can be obtained only when the eccentric path is a straight line and the eccentric rod is of infinite length.

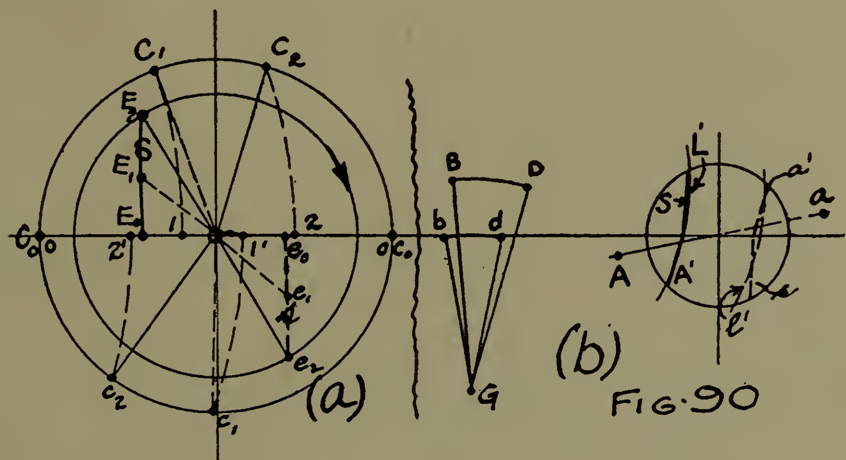


FIG. 90

(c) **Equalization of Cutoffs.** Having first obtained from a Bilgram diagram the throws and angles of advance of the eccentric for the different cutoffs, a diagram like Fig. 90a may be constructed to show the desired positions of the piston, crank and eccentric for the various cutoffs. In this figure, let 2 be the position of the piston for cutoff in the forward stroke and 2′ be that for the return and let O—2=O—2′. Then by drawing the connecting-rod arcs, the corresponding crank-pin positions C_2 and c_2 are found. Next, using the throw and angle of advance of the eccentric (obtained from the Bilgram Diagram) for this particular cutoff, the corresponding positions of the eccentric can be found. Supposing the valve takes steam from the middle, in which case the eccentric follows the crank at an angle equal to (90 deg.—angle of advance), the eccentric positions are found to be E_2 and e_2.

Again, if it is desired to have cutoff occur when the piston is in positions 1 and 1′ respectively, the corresponding locations of the eccentric will be found in a similar manner to be at E_1 and e_1.

If the cutoff is at zero stroke, the eccentric positions at the time of this event are E_0 and e_0.

The lines S and s through the two sets of points (E and e) thus obtained are the loci of the desired positions of the eccentric at the times of cutoff at the two ends of the cylinder.

In Fig. 90b, in which the loci S and s are again drawn, B is the position of the rocker pin for cutoff at the head end, at which time the edge of the valve is even with the port edge, and D is the position of the pin for crank-end cutoff. Then, to have the cutoff occur as desired, it is evident that all

points in S should be at a distance from B equal to the length of the eccentric rod, and that points in s should be at the same distance from D. Evidently equalization throughout the whole range can not be obtained. The best that can be done is to so locate B and D that the eccentric-rod arcs L′ and l′ will coincide respectively with S and s as nearly as they can be made to, but in any case these arcs should pass through the positions of the eccentric for normal cutoff, so that this event will at least be equalized when the engine is operating under normal conditions.

The foregoing method applies equally as well to the case of the direct driven external valve, as to that of the internal valve.

(d) **Equalization of both the Cutoffs and the Admissions.** This can be approximated only when the eccentric follows the crank, as it does when either the direct-driven internal valve, or else the external valve with reversing rocker, is used. Referring to Fig. 90b, A′ and a′ are the respective positions of the path of the eccentric for equal lead-angles at the two ends of the cylinder, the crank positions being A and a. As before, B is the position of the rocker pin when the edge of the valve at the head end is even with the edge of the port, which is the position both for cutoff and for admission, depending on the direction of the motion of the valve. D is the position of this pin for the corresponding crank end events. To have both the admission and the cutoff equalized, B must be so located that the eccentric rod arc L′ will coincide with both A′ and S; and D must be so placed that l′ will coincide with a′ and s. Equalization throughout the whole range is of course impossible, but it can be approximated, and exact equalization for one point of the eccentric in its path, say that for normal load, can be obtained.

(e) **Equalization of Exhaust Events.** Having equalized as nearly as is possible the steam events, one of the exhaust events may be equalized for one particular position of the eccentric in its path, by using unequal exhaust laps, which may be found in the way that was described in § 24.

(f) **General.** Because the valve motion is distorted when an oblique rocker is used, the valve diagrams can be used only to get the throws and angles of advance of the eccentric and the crank positions for the events. On them the laps and openings are merely nominal; the actual values of these can only be obtained after the positions of the valve-rod pin on the rocker has been found, which can be done in the manner that was explained at the end of § 67a.

If the eccentric leads the crank, it is found that the valve openings at the crank end are larger than those at the head. Preferably, however, the head end openings should be the larger, for the velocity of the piston is greater near that end of the stroke than when near the crank end (because of the "angularity" of the connecting rod). If the eccentric is placed behind the crank, in the position it occupies when the valve is internal, or when a reversing rocker is used with an external valve, the larger openings will come at the head end as they should.

Instead of using a bell crank as a guide, there may be substituted an oblique crosshead like that in Fig. 34a, or a simple oblique rocker, with eccentric and valve-rod pins co-axial.

As first laid out, it may be found that the valve openings actually obtained are not satisfactory. These openings can of course be remedied by changing the throw of the eccentric, but if a bell crank is used the ratio between the two arms can be altered so as to produce the desired result. If this change necessitates having the eccentric rod oblique to the center line of the engine, the eccentric path must be shifted to correspond.

68. SUMMARY. In designing a valve gear of this type the following steps would be taken :—

(a) Having determined the maximum valve opening and having assumed the lead, construct a Bilgram Diagram for the latest C. O. (say ⅔ or ¾ stroke).

(b) Assume a path of the eccentric (or of Q) to suit the arrangement of the governor and to give the best steam distribution obtainable (§ 66).

(c) Determine the size of the exhaust laps (§ 62) first so that when the cutoff is the latest there will be sufficient compression to cushion the reciprocating parts ; second, so that when the cutoff is that for friction load, the pressure at the end of compression will not exceed that of the steam ; then use the smaller of the two laps obtained. If it is desired to equalize the exhaust events this may be done for one position of the eccentric in its path, say that for normal load, by using unequal laps which may be found in the manner given in § 22.

(d) To show more clearly the action of the valves than is done by the Bilgram Diagram, the Elliptical Diagram may be drawn (§ 63). If desired, the Zeuner and Sweet Diagrams can also be constructed at this stage (§ 64).

(e) The other proportions of the valve, which is assumed to be of the Sweet type, may then be determined by constructing the Barr Diagram of Maximum Openings as in § 65.

(f) If it is desired to attempt the equalization of the events, this can be done by the "cut and try" methods outlined in § 67.

THE DESIGN OF

VALVE GEARS

FOR

STEAM ENGINES

BY

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